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THE DEVELOPMENT OF MACHINERY IN THE U. S. NAVY DURING THE PAST TEN YEARS.

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INTRODUCTION.

When I assumed duty as first assistant in the Design Division of the Bureau of Steam Engineering of the U. S. Navy Department in July, 1906, I little realized that I was being put in a position where opportunity would be offered me to participate in the greatest advances in marine propulsion that the history of marine engineering has ever known.

During the period covered by this paper we have seen the marine reciprocating engine reach its highest point of development only to be superseded by the steam turbine, which, as in the case of the older engine, was directly connected to the propeller shaft. The direct-connected turbine has had only a short life, and at the present time is rapidly disappearing from

the marine stage, its place being usurped by the mechanical and the electric reduction drives.

Coincident with these changes in the types of steam engines for propelling machinery has occurred the development of the internal-combustion engine as illustrated by that most important of the type, the Diesel engine.

Accompanying the advance in main propelling engines were improvements in boilers, in air, circulating and feed pumps; in forced-draft blowers; the adoption of oil for fuel in place of coal, and the development of a method for the more accurate design of propellers for the conditions under which they were to operate.

It is my intention to take you through the marine engineering history of these past ten years as I have seen it in order to give you in the limited time at my command as clear an understanding as possible of what has been done and of why it was done.

CONDITIONS IN 1906.

Upon assuming my duties in the Bureau of Steam Engineering, of the various vessels under construction there were only five whose machinery plants have had any influence upon design during the period of development of subsequent years. These five vessels were

Battleships—*Michigan* and *South Carolina.*

Scouts—*Birmingham, Chester* and *Salem.*

The two battleships, aside from their interest as the forerunners of the Dreadnaught and Super-Dreadnaught classes of vessel, were of interest from an engineering point of view on account of having the first examples of perfected reciprocating engines for main propelling engines and on account of being the first vessels in the Navy to be originally built with superheaters applied to the boilers.

The scouts were particularly interesting, as two of them carried the first installations of turbine-propelling machinery that existed in our Navy, the *Chester* being fitted with a four-shaft

arrangement of Parsons reaction turbines, while the *Salem* had a two-shaft arrangement of what is known as the Curtis velocity-compound impulse turbines. The third vessel, the *Birmingham,* was fitted with high-speed reciprocating engines in order to give a thorough comparison of the three systems.

Valuable lessons were taught us by both battleships and scouts. These lessons were as follows:

Lessons Learned from "Michigan" and "South Carolina."

Engines.—The engines of these ships were fitted with piston valves of such length that the steam ports were perfectly straight and entered the steam cylinders in the plane of the diameter. This change from the old system, in which the valves were short and the steam passages long and crooked, permitted the use of ports of larger cross-sectional area resulting in a decrease in steam velocity through the ports with consequent reduction in steam-pressure drop between the valve chest and cylinder, while at the same time a very considerable reduction in the volume of the ports and, consequently, in the total clearance volumes was realized. The clearance percentages of the old type of engine, referred to the volumes swept by the different steam pistons averaged about

H.P. cylinder	22½ per cent.
I.P. cylinder	18½ per cent.
L.P. cylinders	16 per cent.

With these first engines of the new type these clearances were reduced to

H.P. cylinder	12⅛ per cent.
I.P. cylinder	12 per cent.
L.P. cylinders	11.8 per cent.

Again, in the old type of engines the cylinder ratios, neglecting clearances, were

$\frac{\text{I.P.}}{\text{H.P.}}$	2.66	say $2\frac{3}{4}$.
$\frac{\text{L.P.}}{\text{I.P.}}$	2.65	say $2\frac{3}{4}$.
$\frac{\text{L.P.}}{\text{H.P.}}$	7.046	say 7.

These ratios gave an expansion of the steam of about 9, and while such ratios of cylinders tend to light weight, they also produce low economy at full power on account of the low ratio of expansion.

In order to increase the economy of the engines, the cylinder ratios of the *Michigan* and *South Carolina* were made

$\frac{\text{I.P.}}{\text{H.P.}}$	2.64	say $2\frac{3}{4}$.
$\frac{\text{L.P.}}{\text{I.P.}}$	3.83	say $3\frac{7}{8}$.
$\frac{\text{L.P.}}{\text{H.P.}}$	10.13	say 10.

This change in ratios increased the expansion of the steam to about 12, and reduced the mean effective pressure as referred to the L.P. cylinder from about 55 pounds to 35 pounds per square inch of L.P. piston area.

As previously stated, the boilers of these vessels were fitted with superheaters. The original estimate of the amount of superheat which could be realized from the amount of surface provided was about 100 degrees F., but on trial the superheat at the H.P. valve chest only reached 86 degrees. As the percentage gain in economy by the use of superheat is roughly estimated at 1 per cent. for every 10 degrees, the superheat gave a gain of 8.6 per cent. in economy.

The actual gain in economy of the engines of these vessels was, however, considerably in excess of this, although the exact amount is not known, the water consumption of the engines not having been measured. The fuel measurements,

however, showed a coal consumption of 1.46 pounds per hour per horsepower of all machinery in operation, which, when compared with 1.775 pounds, that of the *New Hampshire*, one of the last of the vessels having the old type of engine and no superheat, indicates a total gain of 21½ per cent. The difference between 21.5 and 8.6, 12.9 per cent., is a rough measure of the gain due to the changes in the engine proportions and design. This gain will later be shown to have actually occurred, as in later engines the water consumption was actually measured.

OTHER POINTS OF INTEREST.

In laying out the main steam piping of the *Michigan* an attempt was made to eliminate all slip joints by the use of generous bends in the pipes and thus rid ourselves of the trouble due to leaks from such joints. The results obtained were very unsatisfactory, as steel piping of the diameters used, 7 inches to 12 inches, was too stiff to expand freely with the amount of bend that could be installed. It was found impossible to keep the flange joints tight, and the entire system had to be replaced by piping fitted with slip joints.

A departure in the material for the main feed system was also made. In previous ships, the feed piping had been of seamless-drawn copper throughout. On the *Michigan* all feed piping of 5 inches diameter and over, was made of seamless-drawn steel.

Forced-draft blowers.—In regard to these, the two battleships also marked the beginning of an epoch; while preceding vessels had all been fitted with blowers of large diameter and slow speed of revolution, driven by reciprocating engines, on these new ships blowers of a type known as the Sirocco were installed, and these were driven by electric motors obtaining their current from the ship's electric plant. While the old-type fans were run at only about 225 revolutions, the new type ran at about 650, resulting in much smaller, lighter ma-

chines, and by the replacement of the reciprocating engines by electric motors, vastly increasing the durability and the reliability of the system.

THE SCOUTS.

These vessels, *Birmingham, Chester* and *Salem,* are chiefly interesting as marking the definite abandonment of the reciprocating engine for light, high-speed work and the adoption of the turbine in its place, the reciprocating engines of the *Birmingham* having been fitted for the purpose of obtaining a comparison of the economy and reliability of the two types of machinery.

The engines of the *Birmingham* ran at 200 revolutions, those of the *Salem,* which vessel was fitted with Curtis turbines, at 350 revolutions and those of the *Chester* at 502 revolutions, the engines developing 16,000 indicated horsepower for the *Birmingham* and 16,000 shaft horsepower for each of the other two vessels, the steam pressure being 250 pounds per gage.

While the *Chester* was the first example of Parsons turbines in our Navy, this type of turbine had been so thoroughly developed in all its details before being pushed forward as a competitor of the steam engine that the installation fitted to the *Chester* was similar in its arrangements and mechanical details to all subsequent vessels in which this type of turbine was installed. The turbines were arranged on four shafts, with an H.P. on one and an I.P. cruising turbine on the other of the two inner shafts. While later vessels with four-shaft arrangements were fitted with backing turbines on all shafts, the *Chester* differed, and backing turbines were fitted to the inner shafts only.

The *Chester* was a successful ship from the outset, as was the *Birmingham,* and both vessels reflected credit upon their builders.

The experience with the *Salem,* however, was far from satisfactory, and the reasons are not difficult to find. In tracing

them out it will be well to give a short sketch of the progress of the Curtis turbine up to the date of completion of the *Salem*.

The Curtis turbine is of the type known as "impulse velocity compounded." It had been used quite extensively on shore for driving electric generators before the marine type was designed. The type used on shore being built as vertical units, it became necessary to depart from this form to adapt the turbine for use on board ship.

At this early date the theory of this turbine carried with it the idea that as all expansion of the steam took place in the steam nozzles, wherein steam pressure was changed into velocity, there was no change in pressure throughout the blades of any one stage, and that therefore there was no unbalanced steam thrust on the rotor of the turbine.

The earliest marine application of the Curtis turbine occurred on the yacht *Revolution* in about 1902. These turbines were provided with only two stages, and developed shaft horsepower equivalent to 1,800 I.H.P. at 672 revolutions. A Board of Naval Engineers reporting on the performance of these turbines in 1903, stated "This plant, being the first large set of steam turbines installed for screw propulsion in the United States, it seems appropriate to mention that the general performance of these turbines, although themselves built rather in the nature of experimental appliances, was quite satisfactory, and they are well adapted as a type of motor for the transmission of propulsive power." * * * "The points in conjecture with regard to these machines being their efficiency and their capacity to compete in economy with the best marine engines, should be ascertained by more accurate tests, both on land and on board ship, where this is possible. The opinion of the undersigned (the Board) is that such tests are warranted, and eventually will result in the application of a motor for screw propulsion highly satisfactory to the naval service."

Following this report the Southern Pacific S. S. Co. placed

an order with the Fore River Shipbuilding Company for the construction of the S. S. *Creole* to be fitted with Curtis turbines, and in 1905 the Navy Department contracted with the same Company for the construction of the *Birmingham* and *Salem*.

The turbines of the *Creole* and *Salem* were made practically from the same patterns, the nozzle areas of the *Salem* being, however, twice as great as those of the *Creole* on account of being called upon to deliver twice the power. The turbines were built with seven-wheel stages, the drum type of Curtis turbine not having yet appeared upon the scene.

The turbines of both of these vessels were fitted with a type of blading which "hind sight" has taught us was so weak as to be a serious menace to the reliability. The diaphragms between stages were made each in one piece, and to overhaul the shaft packing of any diaphragm required that the turbine shaft be completely stripped up to and including the desired diaphragm. Further, the thrust bearings were made completely independent of the turbine casings, which was a serious mistake, as these bearings should come and go with the turbine as it expands or contracts. Also, in the case of the *Salem*, while the turbines of the *Creole* were made rights and lefts, those of the *Salem* were made rights only, the port turbine being exactly the same as the starboard, but being turned end for end when installed in the ship. This resulted in adding the unbalanced axial thrust of the turbine to the propeller thrust and undúly overloaded the thrust bearing and its support, and has been a continual source of trouble ever since the vessel went into service.

A comparison of the machinery weights of the three scouts will be interesting, both actual weights and weights per actual power developed on trial.

The total weight of machinery installation, including propelling machinery, auxiliary machinery, piping, boilers and fittings, smoke pipes and uptakes, lagging and clothing, flooring, ladders and gratings, fittings and gear, stores, tools and

spare parts carried on board, and pipes, valves, etc., connected to machinery not under cognizance of the Bureau of Steam Engineering was

	Dry.	Wet.
For *Birmingham*	760.82 tons	843.97 tons.
Salem	853.85 tons	908.96 tons.
Chester	735.87 tons	800.69 tons.

The weight per horsepower delivered was

		Pounds.
For *Birmingham*	(I.H.P.)	117.18
Salem	(S.H.P.)	112.68
Chester	(S.H.P.)	89.67

Turning now to the question of relative efficiencies of propulsion, and reducing the I.H.P. of the *Birmingham* to S.H.P. by multiplying by .92, the S.H.P. per knot speed per hour required were as follows

Knots	10	15	20	24	25
Birmingham	92	156	331	563.5	
Salem	97.5	193	380	600	710
Chester	80.?	190	362.5	604	758

These power results are not particularly consistent, due, probably, to the characteristics of the hulls of the vessels and to variations in weather conditions under which the trials were held. There may also have been errors in the torsion-meter readings, especially at the low speeds, where a slight error would produce the greatest percentage error.

Now, the true measure of efficiency of propulsion is not horsepower required for any speed, but is either steam consumption or fuel consumption required to produce the speed. Where oil fuel is used the personal element of the operating force is reduced to a minimum, and measurements of fuel are preferable to measurements of steam consumption, as the first gives a gage on the efficiency of the entire machinery plant,

including boilers, while the latter gages the efficiency of the engines and auxiliary machinery only. Where coal is used as the fuel the personal equation of the fireroom force enters largely into the problem, and therefore measurements of steam consumption are to be preferred in such cases.

The steam consumptions in pounds per hour of the three vessels, for all purposes, on these trials were as follows:

Knots ...	10	15	20	24	24.5
Birmingham..	34,000	66,000	134,000	285,500	
Salem	44,000	84,000	150,000	273,000	300,000
Chester	43,200	76,500	138,500	254,800	284,000

The steam consumptions of the main engines, only, at these speeds were

Knots ...	10	15	20	24	24.5
Birmingham..	21,000	50,400	113,000	247,000	
Salem	29,900	66,700	127,300	238,000	261,800
Chester	24,250	56,200	118,000	218,000	244,000

Pounds of water per hour per S.H.P. (main engines only):

Knots	10	15	20	24	24.5
Birmingham	22.82	21.6	17.06	18.26	
Salem	30.7	23.	16.8	16.50	14.75
Chester	30.3	19.72	16.27	15.03	13.34

Attention is called to the fact that while the total steam consumption with reciprocating engines is decidedly better below about 22 knots than that of either the Curtis or the Parsons turbines, even though the latter are fitted with cruising turbines which are in use from 20 knots down, the economies are about even at 22 knots, and above that speed both types of turbines have a decided superiority. Hence, in cases where high speed and light weight were the prime factors to consider in the design of machinery for a vessel, some type of turbine

machinery was always to be preferred over reciprocating engines. In cases where the speed of vessel was comparatively low, where high speed was seldom called for and where economy of propulsion at cruising speeds was of paramount value, the reciprocating engine was, under the conditions of turbine design existing at this early date, very much to be preferred.

For several years after the adoption of turbines as a prime mover for marine propulsion the turbine designers and builders successfully blinded the public concerning the performances of turbine vessels by comparing the steam per horsepower required by turbines with that required by reciprocating engines, and, as shown by the comparison of these amounts for the scouts, the argument was a very good one if the material data necessary to reduce the figures to figures of pounds of water per knot run were not available.

Leaving these vessels we now reach the starting point of what may be called the modern Navy.

1906–1907.

During the summer of this year work was begun on the plans of our first two Dreadnaughts and for two, afterwards increased to five, high-speed destroyers.

The plans for the Dreadnaught were prepared in competition with outside firms, and, while a nominal speed of twenty knots was provided in the design, sufficient reserve power was installed to insure of twenty-one knots being attained.

When the competitive plans were examined by the Department those prepared by the Department were adopted and work on the contract plans and specifications at once started.

The machinery space provided in these vessels was such that only reciprocating engines or turbines of the impulse or impulse-velocity compounded type, similar to those of the *Salem*, could be installed. The Department plans showed reciprocating engines designed on the same lines as those of the *Michigan* and *South Carolina*, the cylinder diameters as de-

signed being 36 inches, 57 inches and two of 76 inches with a stroke of four feet. The contractors, fearing a lack of power in these engines, later increased the diameter of the H.P. cylinders to 38½ inches, a change which was unnecessary for a speed of 21 knots, but which resulted in considerably higher speed of ship.

The plans also showed Babcock & Wilcox boilers fitted with superheaters, the superheating surface being 10 per cent. of the total heating surface. While in earlier vessels, in designing the boiler installations the empirical rule used had been 15 I.H.P. per square foot of grate surface and a ratio of 44.5 to 1 between heating and grate surface, in the design of these vessels the allowance was 18 I.H.P. per square foot of grate and a ratio of 44.5 to 1 between total heating surface, including superheaters, and the grate.

These Dreadnaughts also marked our first step into the field of oil-fuel burning. When it was decided to fit these vessels with oil fuel the best system to adopt was not known. The entire field of oil burning was carefully gone over and it was determined to fit them with that system which would require the fewest number of changes from the ordinary coal-burning installations, especially as in the case in view coal was to remain the principal fuel, the oil only being fitted as an auxiliary to be used when coal became difficult to trim or in emergencies. This requirement made it necessary to adopt a system with which the forced-draft blowers required by the coal-burning installation could be used, and we were therefore driven to the now well-known system of "Mechanical atomization" as it is called, although "Pressure atomization" would more exactly define it.

In the arrangement of furnaces for burning either coal or oil the grates are fitted as for ordinary coal-burning boilers, and the burners for oil, with the necessary slotted cones for the proper admission of air, are located between the furnace doors. Owing to the limited volume of the furnace between the tube roof of the furnace and the grates, only very limited

quantities of oil can be successfully burned in such installations. The whole question of oil burning will, however, be discussed more fully later on.

ADOPTION OF TURBINES.

Before the advertisements for bids for these two Dreadnaughts were issued the first turbine-propeller battleship, the British original *Dreadnaught,* the name of which has been applied as a general name to the "all big-gun battleship," had her trials, and the results were spread broadcast as a world advertisement for this new form of propulsion. It is spoken of here as a new form for the reason that up to this period it had only been used for much smaller, less important installations in destroyers and in the small British cruiser *Amethyst.*

The installation on the *Dreadnaught* was of the very simplest type, there being four shafts with high-pressure ahead turbines on two shafts and low-pressure ahead and astern turbines on the remaining two. There were no means whatever provided for increasing economy at low speeds, and the expenditure of steam at these speeds was extravagant.

The new type of machinery, however, immediately struck both the public and the Service fancy, and the Navy Department, yielding to the clamor in favor of the new fashion, invited bids for turbine machinery in addition to those for reciprocating engines.

As previously stated, the engine rooms on these vessels were too small to accommodate any other type of turbine than the impulse or the impulse-velocity compounded type. When the bids were received a satisfactory bid was made by the Newport News Shipbuilding and Dry Dock Company for one ship built according to the Department plans and specifications, that is with reciprocating engines, while the Fore River Shipbuilding Company offered a satisfactory bid with guarantees of performance for a vessel with Curtis turbines.

The turbines offered were in general similar to those fitted

in the *Salem,* but, inefficient and troublesome as they proved themselves later in actual service, they did mark one big step in the development of this now highly successful and economical American turbine, and the development of such a successful completed machine, in my opinion, completely justifies the early experiment and failure in the *North Dakota.* The improvement made consisted in increasing the turbine stages from seven to nine, the last four stages being carried on a drum in place of on separate stage wheels. This was the first appearance of drum construction with turbines of the Curtis and similar types, although it was and is the standard method for those of the Parsons type.

Until the advent of the Curtis turbine the Navy Department had been forced to go abroad for its turbine designs, a position which was extremely undesirable and which justified experiment to the fullest extent to relieve us of the situation.

FORCED LUBRICATION.

During the building of the engines of the *Delaware,* reports were received of the results obtained with forced lubrication on the reciprocating engines of the British battleship *Africa.* These reports indicated such a great improvement over the ordinary method of gravity lubrication that it appeared desirable to incorporate it in the engines building. Plans were at once prepared by the Bureau of Steam Engineering and forwarded to the Newport News Company, with a request that the matter be taken up at once and an estimate for the cost of the change be submitted to the Department in order that authorization for the additional expenditure required for this improvement might be obtained. This was done and, after some delay, the necessary authorization was obtained.

The original plans of forced lubrication for the *Delaware* contemplated applying it only to main and crank-pin bearings and to the eccentrics, depending on the original gravity feed for lubrication of crosshead slides, wrist pins and valve-link

bearings, and, in conformity with this idea, only those parts having forced lubrication were covered in by the oil casing.

When the vessel was taken out for her preliminary acceptance trials the crossheads and slippers showed a tendency to run warm, and, before the trials, temporary connections through the water-service piping were made from the forced-lubrication system to these parts. The results obtained due to the forced lubrication were very marked, practically all the vibration and noise to which we are accustomed with the ordinary system of oiling being noticeable by their absence.

The trials, however, demonstrated the desirability of extending the forced lubrication to all the moving parts of the engines, and also that, when extended to the crossheads and slides, a large amount of oil was thrown on the piston rods and lower heads of cylinders, vaporizing on the latter, the oil on the piston rods and the oil vapor being drawn into those cylinders where pressures below the atmosphere existed. This oil then passed over to the boilers and caused fear of trouble due to the oiling up of the water side of the heating surfaces.

DEVELOPMENT OF THE PISTON-ROD "STEAM SEAL."

The carrying of oil to the boilers being considered a serious menace, it was imperative that means to prevent it should be devised as quickly as possible, and in aiming at the preventive advantage was taken of the fact that for cylinders in which steam pressures lower than the atmosphere are apt to occur, the piston-rod outer packing must be designed to resist pressure from the outside. As the M.P and L.P. cylinders are those in which this condition exists, and as the high pressures carried in the H.P. cylinders tend to cause a blowing out of steam along the piston rod, cutting off the oil from the rod and preventing the entrance of oil vapor, it was necessary to apply the preventive to the M.P. and L.P. rods only. The cure finally adopted is now known as the "steam seal," and consists of an outer stuffing box carrying about three rings

of elastic packing with a space between the first or lower ring and the upper two, into which steam of a slightly higher pressure than that likely to exist in these cylinders is admitted. When the regular piston-rod packing is properly fitted and is operating properly the "steam seal" does its work efficiently. When the regular piston-rod packing is not doing its duty the steam seal may aggravate the trouble by blowing through into the cylinders.

In service this system of lubrication proved so satisfactory that little time was lost in extending it to quite a number of the older vessels of the fleet where equally satisfactory results have been obtained. One of the commanding officers of the fleet made the remark to me that we had builded better than we knew, as the reciprocating engine fitted with forced lubrication was quite a different machine from the same engine fitted with gravity feed. Only the other day a report was received in the Bureau from one of the armored cruisers stating that they had just overhauled their crank-pin and main journals after a period of two years' running, and none of the bearings showed the slightest indication of wear. This, in itself, shows the immense decrease in work required to keep these engines in order over that required in the old days before forced lubrication was adopted.

RESULTS OF TRIALS OF "DELAWARE" AND "NORTH DAKOTA."

In the contracts for these two vessels appeared, for the first time, a requirement for the measurement of the steam used, this for the *North Dakota* being accompanied by guarantees that the following steam consumptions per shaft horsepower would not be exceeded:

Full speed	15.1 pounds.
19 knot	16.1 pounds.
12 knot	23.2 pounds.

The *Delaware's* trials took place off Rockland, Me., in October, 1909, and those of the *North Dakota* off the same place in November of that year. The results obtained were:

Ship.	12-knot.		19-knot.		Full speed.	
	Del.	*N. D.*	*Del.*	*N. D.*	*Del.*	*N. D.*
Speed	12.24	12.04	19.217	19.21	21.56	21.64
I.H.P. (main engine)	3,905	.....	16,602	.....	28,578	.....
S.H.P. (main engine)	.....	3,757	.....	16,713	.....	31,862
Coal per I.H.P. or S.H.P.	2.673	2.64	1.795	1.63	1,888	1.71
Water per I.H.P. or S.H.P.	19.37	27.12	13.91	16.67	14.32	14.

In advertising the results of this first large Curtis turbine installation, the advocates of the turbine in comparing the two vessels religiously confined themselves to the comparison of coal per horsepower, and always called your attention to the fact that a shaft horsepower was only 92 per cent. of an indicated horsepower, and that, therefore, the fuel consumptions were:

	12 knot.	*19 knot.*	*Full speed.*
Delaware	2.9 pounds.	1.95 pounds.	2.05 pounds.
North Dakota	2.64 pounds.	1.63 pounds.	1.71 pounds.

per shaft horsepower, thus indicating the vastly greater economy of the steam turbine and its therefore much greater superiority as a marine engine.

They carefully neglected, however, to call attention to the following comparison, which gives the measure of the relative propelling efficiencies of the two types of machinery:

	Total pounds of water.		
Speed.	*12 knots.*	*19 knots.*	*Full speed.*
Delaware	83,463	243,810	422,931
North Dakota	101,883	281,784	445,979

These figures, which glaringly demonstrated the superiority of the reciprocating drive as compared to the then current tur-

bine practice, have been maintained and even increased by the *Delaware* in service as the efficiency of the *North Dakota's* turbines rapidly decreased while at the same time the turbines became unreliable, finally necessitating her withdrawal from active service with the fleet until new machinery could be built and installed.

DESTROYERS OF 1906.

No torpedo boats or destroyers had been appropriated for since the completion of the first lot of sixteen destroyers, which had occurred in 1901–1902, these having been authorized in 1898 and having taken from three to four years for completion.

These original destroyers were of about 7,500 I.H.P. and were designed for 28.5 knots trial speed. After many difficulties and a reduction of trial speed to 27.5 knots, they were finally squeezed through and accepted. Great troubles were experienced with their light and fast-running reciprocating engines, and with their forced-draft blowers, which were also driven by reciprocating engines.

In 1906 and 1907 five more destroyers were authorized. The estimated I.H.P. for this speed was 10,000, and the Navy Department prepared plans showing Normand reciprocating engines for this power. In advertising for bids for the construction of these vessels bidders were invited to submit plans showing turbine machinery, both with and without cruising turbines, and were given to understand that, other things being equal, turbine installations would be preferred.

These vessels, as finally contracted for, were all of three-shaft Parsons turbine type, with a H.P. and I.P. turbine, one on each outboard shaft, for cruising purposes. The vessels were all coal burners, and while, during their construction, it was proposed to change to oil fuel, the cost of change and the difficulties in the way due to progress of construction were found to be so great that the idea was abandoned.

There never was any doubt in the mind of the Department

as to the desirability of displacing the unreliable high-speed engines that had been previous practice for these light high-speed vessels by the more reliable turbine, and the step was taken with no hesitation. There was also another step in advance made at the same time, and this step was even of more value, if possible, than the adoption of steam turbines for the main propelling engines.

STEAM TURBINE-DRIVEN BLOWERS.

One of the great sources of failure to hold high speed for any long continued period of time with all classes of vessels had been the continually recurring break-downs of forced-draft blowers with engines of the reciprocating type. These had been done away with, and blowers driven by electric motors substituted for battleships, beginning with the *Michigan* and *South Carolina*, as already stated.

In two of these five destroyers appeared, for the first time, the turbine-driven forced-draft blower. In laying down the plans for these vessels the Bureau of Steam Engineering had scanned the market closely to find turbine-driven blowers of proper design to install for the work required. Not being able to find any such in existence, it invited in the specifications the installation of such type in place of the type directly specified. Meanwhile it succeeded in awakening the interest of the Terry Steam Turbine Company in the proposition, and by the time the contracts for the construction of the vessels were awarded, this Company had developed what appeared to be, on paper at least, a very satisfactory unit.

The Cramp Ship and Engine Building Company were quick to see the advantages in this new unit and proposed its installation in the two vessels building by them. This proposal was approved and the units installed. The contractors for the remaining three vessels, however, adhered to the old reciprocating-engined blowers, arguing the greater steam cost with the small, inefficient turbines, losing sight, however, of the

fact that with feed heaters installed and with turbine main engines in which excess auxiliary steam can be used in the heaters and in the low-pressure stages of the turbines, steam economy of auxiliaries only used when the main engines are in operation or used to only a small extent in port, does not cut any great figure in the general economy of the machinery plant. Also that the greater reliability of the turbine blowers would, if realized, more than offset any possible loss in economy, as the forced-draft blowers, in rate of importance to a destroyer, are as necessary as a good heart is to the human body.

The blower units as fitted were of the vertical, direct-connected turbine type, with single-stage Terry turbines driving what are known as multivane fans of the Sirocco type. These fans are located directly at the base of the intake ventilators and discharge throughout their complete circumference into what is called a diffuser chamber, in which the velocity of the discharge becomes reduced and the air pressure after discharge from the fans increased.

The diffusion chamber is formed by two plates forming frustrums of cones with the bases separated several inches at the outer circumference; the inner edges of the plates at the fans being separated to the width of the fan blades.

When the vessels carrying the new type of blowers were tried great trouble was experienced due to two fans discharging into the same fireroom; unless the fans were running at practically the same speeds, that fan which was doing the least work would be wrecked on account of the air being forced back through it. After considerable trouble and experiment the cause of the trouble was located. It was due to the too great width of discharge opening from the diffuser chamber. This error was corrected and the fans by their later performances installed such confidence that no other type of blower than that of the turbine driven has since been considered for destroyer work, and in later vessels of other types it has even

shouldered the electric-driven fan out of the way, as the flexibility of control is so much greater.

EXPERIENCE WITH OIL FUEL ON THE "DELAWARE" AND "NORTH DAKOTA."

The oil-fuel handling of these two vessels varied in one respect, and that was in the location of the settling tanks, that of the *Delaware* being located aft alongside the engine room, while the *North Dakota* followed the Bureau plans and had a small cylindrical tank located in the upper part of each fireroom. This location on the *North Dakota* was later credited with a disastrous fire on that vessel in which several lives were lost. The real cause of the disaster was, however, the fact that each of these tanks was fitted with a glass gage for indicating the height of oil in the tank, and oil vapor having collected on the flat tops of the B. & W. boilers, it became ignited in some unknown manner, the flash resulting broke the gage glass and, the booster pump supplying oil to the tank being running at the time, the oil was forced out and burned like a gigantic torch until the pump was stopped and the tank completely emptied.

Orders were immediately issued by the Department prohibiting the use of gage glasses on oil-fuel tanks unless such glasses were fitted with spring valves that were open only while readings were being taken, the valves closing automatically when released.

Other troubles were experienced with the oil fuel due to the inexperience of all hands concerned with it, including the makers of the oil-fuel apparatus. The burners as originally supplied were of sufficient capacity to deliver the full power condition, and this, on account of insufficient volume of furnace above the grates, resulted in heavy carbonization on the grates and bridge walls and in very heavy smoke formation. All manner of criticism was levelled at the system employed, and it was only by taking a strong stand on the subject that

the Bureau prevented it from being superseded by air or steam atomization. The succeeding years have demonstrated the wisdom of the step and we hear no more of propositions to use any other system.

1907–1908.

The next naval appropriation bill provided for two battleships of the Dreadnaught type and for ten destroyers.

The battleships, later named the *Florida* and *Utah*, were very similar to the *Delaware* in hull lines, but of about 1,000 tons greater displacement, requiring a slightly greater tow-rope power at low speeds and slightly lower tow-rope power at 21 knots than was required for the previous vessels.

In laying down the machinery arrangement plans, the fire-room arrangement of boilers and piping was practically duplicated, the boilers, however, being slightly increased in heating surface over the total heating and superheating surface of the earlier vessels, while no superheaters were installed, the Parsons marine turbine not being adapted to use superheated steam on account of the abnormally small radial blade clearances which must be provided in that turbine in the high-pressure stages in order to preserve, or rather realize, a good efficiency.

Following the British lead, the engine rooms were laid out showing a four-shaft installation of Parsons turbines with H.P. turbines, both ahead and astern, on the outboard shafts, and L.P. ahead and astern turbines on the inboard shafts. The inboard shafts likewise carried cruising turbines, a H.P. turbine on one and an I.P. turbine on the other, so that at low speeds the steam would just pass through the H.P. cruising, then the I.P. cruising, thence to both H.P. mains, finally through both L.P.'s to the condensers. At intermediate speeds it was admitted to the I.P. cruising direct, the H.P. cruising running in vacuo. At higher speeds both cruising turbines were cut out and ran in vacuo.

NECESSITY FOR CRUISING ECONOMY IN OUR BATTLESHIPS.

When the conditions under which our battleships must operate are compared with those which faced European nations in the period directly preceding the present great war, a war which they all evidently had in mind in the preparation of the machinery designs for their vessels, it is readily understood why they could afford to neglect any provisions for insuring great cruising radius, while with us the necessity for providing the maximum cruising radius is imperative.

The former nations were providing for operations in confined waters such as the North Sea and the Mediterranean, where they were never far from their bases. The runs from base to station were short, and occasion might arise where these runs would have to be made at high speed. Refueling became a simple proposition. Under the conditions foreseen high speed was more desirable than cruising radius, and the latter was sacrificed.

Turning now to the conditions confronting us, we see the entire western half of the North Atlantic ocean from the northern shores of South America to Greenland on our Eastern sea front, and the entire line from Seattle to Panama to the southward, and from Seattle to Honolulu, thence to Guam and on to the Philippines and still further to Samoa, requiring our attention. The areas to be covered are great and the distances to bases and from base to base in some cases are magnificent. Fuel economy is of the highest value, even predominates over high speed, as the refueling problem becomes a serious one and the greater our bunker capacities and the greater the fuel economy of our machinery, by so much is the seriousness of the problem reduced.

It will be shown later how this question had an important bearing on the decision of the Department as to the types of machinery to be fitted in some of our subsequent vessels.

Beyond the installation of Parsons turbines in these two vessels the only point of practical engineering interest was a

change in the basis of design of feed heaters. The old empirical rule used for years in the design had been to allow one square foot of heating surface for every ten designed I.H.P. Such a rule took no account of the amount of heating medium, which in the case of naval vessels is the auxiliary exhaust steam, the temperature of the heating medium, nor of the amount of feed water to be heated. During the year preceding the designing of these vessels the Bureau of Steam Engineering had developed a feed heater which, when constructed, had been thoroughly tested out at the Engineering Experiment Station under conditions of exhaust pressure and temperature corresponding to those used in service. From the results of these tests design curves were prepared, and from these curves the necessary surface for the feed heaters was obtained.

In connection with these tests comparative tests were made with and without whirler strips in the tubes, these whirler strips being nothing but ribbons of sheet copper twisted to a pitch of 6 inches and inserted in the tubes, extending their full length. These whirlers, by causing every particle of water flowing through the tube to be brought into intimate contact with the heating surface, increased the efficiency of the heater about 25 per cent.

The heaters developed from the design curves required a total heating surface for both heaters, inclusive, of 700 square feet, while those of the preceding vessels, under the old empirical rules required 2,100 square feet.

In service the new heaters, paradoxical as it may seem, have done much better work than the old ones, delivering the feed at considerably higher temperatures and eliminating the constant troubles of leaky joints that had formerly been experienced.

I am giving the above as an illustration of the beneficial results that may be expected, both in small and in large problems, by carrying out a series of well thought out experiments, and am happy to be able to say that this is only one of the many instances where the Engineering Experiment Station

has demonstrated its value to the Bureau of Steam Engineering and through it to the naval service in general.

Of the ten destroyers two were fitted with Curtis velocity compounded turbines on two shafts, two with Zoelly turbines of a similar type, and six with Parsons turbines on three shafts. The Curtis and Zoelly turbines were newcomers in the destroyer field in this country, but from reports received at the time, the Zoelly turbine was in successful operation in the Germany Navy. When these vessels were completed and put into active service, those with the Curtis and those with the Parsons turbines gave satisfaction from the very start. In a short time, however, the vessels fitted with the Zoelly turbines demonstrated that either our information from abroad as to the success of this type of turbine was incorrect or that the American builders did not understand its construction sufficiently well to build a good turbine. Both vessels gave trouble, but while these troubles appear to have been practically conquered in one of the vessels, in the other the turbines went from bad to worse and are now being replaced with machinery of a different type.

The Navy Department, with these vessels, definitely adopted oil as the approved fuel for destroyers. The system adopted being that of "Pressure Atomization," commonly, though erroneously, known as "Mechanical Atomization." The air for combustion was supplied by turbine-driven blowers of the open type, this type of unit having come to stay.

These being the first installations of all oil fuel, they became the school of experience from which many practical points in handling oil fuel were learned. Among these may be mentioned the following as being practically basic necessities for successful fuel-oil burning:

1. Steady pressure in oil-fuel supply lines to burners.

2. Burners which will efficiently atomize the fuel and, for boilers of the express type, throw a long flame.

3. High and steady air pressure in the firerooms, which prevents leakage of unburnt gases through the boiler-casing joints

into the firerooms and also provides a life insurance for the fireroom force by preventing flare backs of flame and blow backs of steam in cases of tube rupture.

4. High velocity of air entering furnaces to insure intimate mixture with atomized oil throughout length of flame, thus preventing wreckage of furnace linings by heavy pulsation.

5. Efficient and large areas of fire-brick furnace linings, well secured, to prevent radiation losses and to increase furnace efficiency.

6. The absolute necessity of so locating oil burners that no unburnt oil can come in contact with boiler tubes or fire-brick lining of furnace if smokeless burning is to be obtained and if the building up of heavy carbon deposits is to be avoided.

7. Ample air supply to insure an excess of air when running at full power, as this excess is necessary to insure smokeless combustion.

8. Avoid relatively high gas pressures in the boiler passes and uptakes, as such pressures render furnace conditions unstable and tend to produce pulsation with consequent destruction of furnace linings.

MAGAZINE REFRIGERATION.

The Dreadnaughts up to this time authorized or building were of the five-turret type with the magazines for one turret located between the engine and firerooms. In order to reduce the naturally high temperature of this magazine some system of artificial cooling was necessary.

Two systems have been tried out, one with closed air circulation and the other with the cooling medium circulating through pipe coils in the magazines.

Difficulty was experienced with the first system when first applied, due to the moisture in the air frosting up the air coolers through which the refrigerated air from the machines circulated. These cooler pipes would become heavily coated with frost, which is a most excellent non-conductor, and when

so coated no further degree of cooling of the magazine air was possible. The persons operating the machines did not appreciate this fact, so, as the frost gathered, the machines were driven harder and harder until something would break.

Reports were immediately made laying all the troubles to "defects in design," when really the trouble was due entirely to operation. This criticism of "defects in design" is met with constantly whenever any troubles are experienced. The personnel are always held blameless, and quite naturally, as those operating the machinery are doing so to the best of their ability and are loath to admit that the fault might possibly rest with them.

Following these ships, appropriation was made for the *Arkansas* and *Wyoming* and for eight destroyers. The plans prepared for the two battleships were practically duplications of those for the two preceding vessels.

The Department did, however, invite bids on a layout of combination machinery developed by it. This differed from the ordinary type of combination machinery that had been used in the merchant marine. In this latter machinery the reciprocating engines and the turbine combined with them each developed about the same power, the steam pressure at the turbine inlet being that which is usually met with at the exhaust of a triple-expansion engine, about 12 pounds absolute. With the ordinary merchant vessel the ranges of speed and power used in actual service is very limited, and the machinery can be designed for maximum efficiency within these ranges. With vessels of war of high or moderately high speeds the range between ordinary cruising speed and power and full speed and power is extensive, and the combination machinery must be so designed that the turbine will deliver power down to and, if possible, slightly below the cruising speed in order that it may not cause a drag and an absolute loss in efficiency. To accomplish this it is necessary to design the reciprocating engines as compounds, but having the same ratio between the high and low-pressure cylinders as exists in the ordinary triple-

expansion engine. The function of the low-pressure cylinder is then performed by the highly efficient low-pressure turbine. The initial pressure at the turbine at full power in such a case should be about 30 pounds absolute. The whole scheme really can be considered as one in which the comparatively inefficient H.P. turbine and L.P. cylinder are replaced by the much more efficient H.P. and I.P. cylinders and L.P. turbine.

Bids were received for such an installation, but it was decided not to go into the experiment at that time. About two or three years later, the French Government did install a similar arrangement in several of their new battleships. With this same year, 1909, the subject of electric propulsion first made its appearance on the naval stage. Mr. W. L. R. Emmet approached the Navy Department with proposals to fit electric propelling machinery on one of the new vessels, but the Bureau of Steam Engineering advised against it, considering that the new problem should be at least partially solved by installing and trying out an installation on a less important vessel before committing itself to its installation on a capital ship. The opportunity to do this did not offer until two years later.

The destroyers of this program were all of the two-shaft design. Four of them having improved Zoelly turbines, two, Curtis turbines, and two, Parsons turbines. The latter having a H.P. turbine on one shaft, a low-pressure on the other, and having only a single condenser, which constitutes a very weak point in these vessels.

These destroyers are the starting point for high cruising economy in destroyers, the General Board of the Navy having required that they should have a certain specified cruising radius at 15 knots speed.

This requirement necessitated other than ordinary cruising turbines to realize the necessary economy and, therefore, as the speed of rotation at 15 knots was comparatively low, small compound reciprocating engines working in combination with the turbines were fitted, connected to the main shafting by disconnecting couplings. These engines were disconnected for

speeds above 16 knots. The combination was an unqualified success so far as the resultant economy was concerned, a reduction in fuel of from 25 to 50 per cent. below that required for the main turbines only was realized, 49 per cent. at 10 knots, 37 per cent. at 13 knots and 25 per cent. at 16 knots, results well worth the expense and slight complication involved.

Some of these engines were made reversing, but the majority were made non-reversing, which was the form preferred by the Department. Barring some slight troubles, everything worked well with the exception of a form of friction clutch which had been fitted on four of the vessels.

The history of this clutch is a very good example of the difficulties the inventor encounters in the development of a new idea before success is finally realized, and, being such, I will detail this history to you.

The clutch was oil-controlled of the single friction-disc type, and was designed by Mr. John F. Metten, Chief Engineer of the Cramps' Shipbuilding Co., in an honest and finally successful endeavor to meet the Department's specifications. These called for the small cruising engines to be coupled to the main shafts by couplings which could be connected and disconnected with the machinery in motion.

The clutch, as already stated, was of the friction type. The single friction disc of plate steel was thickened into a boss at the center and bolts passed through this boss and a boss in the end of a short length of intermediate shafting fitted between the after end of the engine shaft and the forward end of the main shaft. The after end of this intermediate shaft was forged out into a Tee end, which was held up to and worked in a slot across the end of a disc on the main shaft, thus providing a flexible coupling to care for changes in alignment.

When the couplings were put into service the first troubles were experienced with the coupling disc, which would heat up under load and warp. To obviate this trouble the edge of the disc was divided into segments by slitting it, the slits cutting in radially for about four inches. The remedy was effective.

The next weak point to develop was the bolting of the disc to the shaft. These bolts were then made much heavier. The shafts then began to break at the point where the form changed from cylindrical to the Tee. This form of flexible coupling was then abandoned and a flexible coupling with entire circumferential engagement was fitted. Everything but the cylindrical portion of the shaft having had its inning and been corrected, the body of the shaft began breaking up badly and some of the shafts broke. The trouble was analyzed as due to torsional vibration and the shafts were at once discarded and replaced with new ones of heavier weight and larger diameter. Since all of these changes were made the clutches in all their parts have been satisfactory.

This year's appropriation also called for one collier. This vessel, the *Cyclops,* was fitted with reciprocating engines of the straight-port type, and, when finished and tried out, gave a very creditable performance. She is mentioned here because a sister ship was laid down the next year fitted with mechanical reduction gear, and the year following, a similar vessel fitted with electric propulsion. These new forms will now be taken up.

MECHANICAL REDUCTION GEAR.

For several years anterior to the date of which I am speaking Sir Charles Parsons, in England, and Rear Admiral George W. Melville, coöperating with Mr. John H. MacAlpine, under the patronage of Mr. George Westinghouse, had been investigating the possibilities of mechanical gearing for interposition between turbines and propeller shafting in order to combine the high efficiency of the steam turbine with the high efficiency of slow-turning propellers. This was not the first appearance of such gearing, however, as it had been used in a few instances years before, but for a quite different purpose. The earlier reciprocating engines were very heavy, of long stroke, and very slow moving, and the gearing in these cases was interposed to bring the revolutions of the propellers up.

Sir Charles Parsons already had an installation in successful operation on the merchant ship *Vespasian.* Mr. Westinghouse had built a unit and had carried out a very successful shop test, but as yet had made no marine installation. The Maryland Steel Company, which received the contract for the *Neptune,* the sister ship of the *Cyclops,* proposed to the Navy Department that the main propelling machinery be changed to turbines and reduction gear and the Bureau of Steam Engineering recommended approval of the proposition.

As the installation was experimental, the contract was so drawn that in case of failure the turbines and reduction gear would be replaced by reciprocating engines similar to those of the *Cyclops,* and in order to allow of this the shaft lines and dimensions were so fixed as to allow the one type of machinery to be removed and the other installed without change of line shafting.

The Westinghouse gear as originally designed had the pinion bearings carried on an I beam, which provided a certain amount of flexibility for the pinion shaft and was supposed to take care of any small disalignment which might occur between pinion and gear wheel. Before the gears were built, however, this I beam was replaced by two hydraulic cylinders to which the oil was pumped by the bearings and which by the oil pressure shown gave an accurate gage of the power being exerted.

Mr. Westinghouse, being before everything else an inventor, saw here his opportunity and allowed his inventive genius to run riot. He designed a new type of marine turbine, the first stage being an impulse one, the remainder being of the reaction type. The big difference between this turbine and the ordinary Parsons turbine was in the location of the backing turbine. This was in the same casing with the ahead turbine, but was directly forward of the H.P. end of that turbine, being steam-sealed from it by a dummy fitted with labyrinth packing. The backing turbine exhausted through the rotor drums to the regular exhaust outlet. This turbine was a failure from

the beginning, as there was continual loss due to steam leakage from the ahead to the backing side, causing both loss in steam and in power. He designed a bridge control on the air-brake principle by which the main engines could be operated from the ship's bridge. This never won favor, and but recently a request has been received from the vessel to remove it. He installed Westinghouse-Leblanc water-sealed rotary air pumps. These pumps used salt water as the sealing medium and discharged it overboard. The loss of fresh water due to the vapor which became entrained with this salt water and was then discharged overboard was enormous. Everything about the installation which had been installed by the Westinghouse Company was a failure with the exception of the reduction gears, which were unqualifiedly successful.

Later the turbines were removed and replaced with others of the same type but greatly improved design; the air pumps were replaced with others using fresh water on a closed circuit as a sealing medium; and in order to improve the efficiency of propulsion, the gears were changed in order to reduce the propeller revolutions from 132 to 110 at full power. The new installation is now on the vessel and, after passing through a few vicissitudes, is operating satisfactorily.

Let us now turn to a more pleasing picture. I refer to the installation of electric propelling machinery on the collier *Jupiter*. This installation was undertaken by the General Electric Co., the year following the beginning of work on the *Neptune*.

This being also an experimental installation the Navy Department guarded itself from loss exactly in the same manner as had been done on the *Neptune*. This necessitated the installation of only one main turbine and generator in order that the shaft lines might not be disturbed.

The generator turbine was of the G. E. Curtis type. The generator is totally enclosed and the rotor has fans on each end which take air in at the bottom of the casing and blow it out through the windings, passing thence through a discharge

duct on top of the generator casing. The exciting current for the revolving field is supplied to slip rings on the generator shaft by one of the ship's lighting sets.

The motors, one on each shaft, are of the induction type, with a stator 120 inches in diameter and a rotor 110 inches. They are located in watertight pits which can not be filled by water from below. Being situated directly under the engine-room hatches they are liable to be deluged with water from above, and the insulation of the windings is therefore made waterproof and are intended to give safe running when partially submerged.

The first turbine constructed for this job was not entirely satisfactory to the General Electric Company and they voluntarily scrapped it and constructed a new turbine having very greatly improved blading. The results obtained with the new turbine showed much improvement, and the completed units were installed. Barring the coming adrift of an internal turbine bolt, the installation has been perfectly satisfactory since first tried out. It has been subjected to water, insulation has been injured so that a spark jumped from one end of the armature to the other, the ship has gone through exceptionally heavy weather, and the installation is as satisfactory today as when first installed.

The next vessels to be contracted for, that is, battleships, were the *Texas* and *New York*, and the Navy Department was accused of having taken a step backward when it was decided to put reciprocating engines in these vessels. The reasons for this step will now be given. A careful study of the performance of the *North Dakota* and a better understanding of the factors governing propeller performance had assured the Bureau that no great economy of propulsion could be expected so long as the turbine builders adhered to what was at that time current turbine practice. That is, as pointed out before, they were sacrificing propeller efficiency deliberately in order to obtain a high turbine efficiency and so be able to hold down weight and space to the lowest possible limit. The Bureau,

by its action, was attempting to force them into a change by which the greatest combined efficiency of turbine and propeller could be obtained, and, as later developments indicated, these efforts finally met with success.

In furtherance of this idea, while the plans as prepared showed an installation of Parsons turbines, the Bureau unofficially informed some of the prospective bidders that a proposal to fit reciprocating engines would be given serious consideration.

When the bids were opened the Newport News Company was found to have submitted such a proposal and the Bureau of Steam Engineering promptly recommended acceptance. This action of the Bureau had not yet become public when parties interested in the Curtis turbine invaded Washington and began a vigorous campaign for that turbine, at first urging the installation of turbines similar to those they were then installing on the Argentine Battleships *Rivadavia* and *Moreno,* turbines which had to be nearly completely rebladed with a heavier type of blade than originally installed before they could even get through their preliminary acceptance trials. When they did finally pass these trials the water-consumption trials at all speeds were far in excess of the guarantees. Later in the campaign the advocates of the Curtis turbine proposed the installation of compounded turbines, that is, a high and a low-pressure turbine on each shaft, the installation to be a two-shaft one. The fight was taken up by the Bureau and was carried to the General Board, which was made the judge in the case by the Navy Department. No guarantees were too severe for the turbine people to meet, on paper, so long as they could equalize the estimated performance of the reciprocating engines. Fortunately the Bureau was the victor, and reciprocating engines were installed in the two vessels now known as the *Texas* and *New York,* and the Navy Department has never had occasion to regret this action. It is pertinent here to call attention to the fact that some of the parties engaged in this controversy on the turbine side are now even

more actively engaged in a newspaper campaign to force turbine-driven reduction gear into the battle cruisers which we are just starting to build, and in this case every bit of evidence they have to support their contentions as to the superiority of the installation they are advocating over that of the electric drive is hearsay of the most meagre description. These same parties were also strongly interested in the designs of machinery for the *North Dakota, Salem, Henley, Duncan* and *Downes,* designs which have all proved unsatisfactory and which either have been or are being replaced by machinery of different types or of greatly modified design.

As stated before, the Department has never had occasion to regret its decision in regard to the type of machinery installation for the *Texas* and *New York,* for from the coal-consumption reports received from the fleet, it costs no more to drive the *New York* at 12 knots than it does to drive the *Delaware,* and 17 per cent. less than the *Florida* and *Utah,* and 27 per cent. less than the *Arkansas* and *Wyoming,* even with the much heavier displacement of the *New York* than that of the three vessels first named and the slightly less displacement of the *Arkansas* and *Wyoming.*

Turning again to the destroyers, the next step forward in the design of their machinery consisted in the adoption of turbines and reduction gear for cruising units, these replacing the small reciprocating engines which had been fitted in the preceding classes. This change was brought about by the General Board, which raised the speed for the guaranteed cruising radius from 15 up to 20 knots. As this latter speed required too high revolutions for a reliable reciprocating engine, it became necessary to look for something that was reliable to replace it. The turbines with reduction gears was the answer.

To carry the destroyers up to date and clear them off the board, I will carry them through their remaining history at once. The next and only change which has occurred with their machinery since the adoption of reduction gear for cruising has been to pass from the direct-connected turbine

main drive and adopt turbines and reduction gear for this purpose. This action led to the abandonment of special cruising elements for destroyers as quite a good, (and nearly as good as with the cruising elements), economy at crusing speeds could be obtained with the main turbines, while the economy at high speeds was enormously increased. The weights of the six types of destroyer machinery, based on designed power, are as folows:

Main Propelling.	Cruising Element.	Total weight lbs. per designed S.H.P.
(1) Parsons turbines.	Cruising turbines.	51
(2) Curtis turbines.	Cruising nozzles.	56
(3) Parsons turbines.	Cruising engines.	47.5
(5) Curtis turbines.	Cruising engines.	48
(6) Parsons turbines.	Reduction engines	46
(7) Curtis turbines.	Reduction engines.	46.5
(9) Geared turbines.	.....	42

As the developments of battleship machinery for the next two programs were intimately connected, these two programs will be discussed together, so far as main propelling engines and provision for cruising economy are concerned.

Taking up another very important change at first we will turn to the question of fuel. Experience with oil fuel on the destroyers having been so highly satisfactory, after thoroughly going over the question, the Navy Department decided to fit the future vessels of the first line to burn oil fuel exclusively. The reasons governing this decision are:

(1) Greater ease in regulating steam supply;
(2) Less wear and tear on the boilers;
(3) Increased boiler efficiency;
(4) Maintenance of boiler efficiency;
(5) Decreased complement of men;
(6) Less manual labor required in operation and maintenance;

(7) Greater regularity in steam pressures;

(8) Less hull deterioration from corrosion;

(9) Greater ease and less interference with routine of ship, in and while refueling;

(10) Decrease in fuel weight for given cruising radius;

(11) Decrease in total bunker capacity;

(12) Ability to use otherwise waste spaces for carrying fuel;

(13) Ease with which fuel can be delivered to the boilers.

In opposition to these advantages there are only two disadvantages:

(1) Greater cost of fuel due to possible limited supply to meet all demands, commercial as well as naval;

(2) Less general distribution of oil over the globe necessitating the transportation of fuel to distant points where oil fuel may not be available.

The advantages to be gained far outweighing the disadvantages, the step was taken and oil fuel was adopted. Other nations have claimed the honor of having led in this path, but their claims are not justified. Our *Oklahoma* and *Nevada* were the first sea-going battleships ever designed for the use of oil as the sole fuel.

When this determination was made it was determined to specify boilers of the large-tube express type, that is, generally similar in design to destroyer boilers except the diameter of the tubes which was increased from 1⅛ inches and 1¼ inches up to 1⅝ inches, as the Babcock & Wilcox boiler as designed, and which had been practically adopted as the standard battleship boiler, was not considered suitable for oil-fuel work on account of the low angle of the tubes, which was only fifteen degrees from the horizontal, and on account of the metallic side water walls of the furnace.

Having so decided, superheaters, while very desirable, were omitted, as experience has demonstrated that superheaters fitted in uptakes outside the boiler lack durability, their deterioration being much more rapid than that of the boiler.

The *Nevada* was built with this type of boiler, the Yarrow. The Babcock & Wilcox Company had, however, in the meantime given way to the wishes of the Bureau of Steam Engineering and had redesigned their boiler, replacing the water walls of the furnace with fire brick, increasing the angle of the two lower rows of tubes next the furnace, which were 2 inches in place of one row of 4-inch tubes, and which absorb the maximum amount of heat, to 20 degrees, while the remainder of the tubes were angled at 18 degrees. In addition, the boilers were fitted with downcomer pipes from the ends of the steam drums to the outer ends of the front cross boxes.

After these changes in design were made the Bureau felt assured of the success of the new boiler, and when the contractors for the *Oklahoma* proposed its installation in that vessel, favorable action was recommended to the Navy Department and the new type was installed.

During the preceding few months a plant for testing out boilers under oil fuel, for experimenting in oil-fuel burning and for instructing men in the art of oil-fuel burning, had been erected at the Navy Yard, Philadelphia. In this plant had originally been installed two boilers, Normand and Yarrow, of the large-tube battleship type, and the first boiler of the *Oklahoma* type to be completed was purchased and installed in the plant. Comparative tests of the three boilers showed that the B. and W. possessed a marked superiority in economy over the Yarrow while the Yarrow was somewhat superior to the Normand.

The propelling machinery shown and specified was of the reciprocating-engine type, fitted with forced lubrication and with two condensers per engine, one located at each end of the engine with a short, large-diameter connection to the adjacent low-pressure cylinder. These exhaust connections were fitted with accordion joints to care for expansion and vibration, and with gate valves by which the condenser could be cut off from its cylinder. In each L.P. receiver pipe was fitted a gate valve by which, in case of a badly leaking condenser or disarrange-

ment of a L.P. cylinder, the cylinder involved could be cut out and the engine run as a three-cylinder triple, at reduced power. Each condenser had its own air pump and circulating pump. The air-pump channel ways were all cross-connected in order that any number of the pumps could be used as desired. The two circulating pumps at the same engine ends had their suctions cross-connected. This enabled suction to be taken from either side of the ship in case of grounding. It also permitted suction of one pump to be taken through the condenser on the opposite side of the vessel, using the outboard delivery of that condenser as a main injection. In fact every possible measure was taken to render the ship as immune from breakdown.

While the designs were being prepared the Fore River Shipbuilding Company appeared on the scene with a proposed layout of turbine machinery consisting of a high and a low-pressure turbine on each shaft, there being two shafts, and with the revolutions decreased from the 280 of the *North Dakota* to 220. An estimate of performance of this proposed installation being prepared by the Bureau, it was found that it would require approximately the same S.H.P. for the different speeds as the I.H.P. for the same speeds of the reciprocating-engine ship. This was a large reduction in power as compared with previous turbine-driven vessels, and the Bureau recommended that this plan be accepted, requiring, however, that additional changes be made to give better economy at cruising speeds and securing the guarantees by abnormally heavy penalties. The means provided to obtain this economy a cross-connection was to be provided between the exhaust pipes of the high-pressure turbines on the two shafts in order that, at reduced powers, steam could be taken in the high-pressure turbine on one shaft and exhausted into the L.P. turbine on the other, the other two turbines running in vacua. This doubled the amount of steam flow through each active turbine and so held up the economy. This fitting was never installed, how-

ever, as a much better scheme was developed with the next year's battleships.

The contracts as carried out provided one battleship, the *Oklahoma,* with reciprocating engines and Babcock & Wilcox boilers, and one battleship, the *Nevada,* with compound Curtis turbines on two shafts, with turbines and reduction gear for cruising. Neither vessel was fitted with superheaters.

The change in the cruising apparatus came about through action taken on the design of propelling engines for the battleships next appropriated for, the *Pennsylvania* and *Arizona.*

During the preparation of plans for these two vessels, the Bureau of Steam Engineering was approached and inquiries made as to whether turbine installations would be considered. The inquirers were told that if turbine installations promising an equal or nearly as low a fuel expenditure as the reciprocating engines shown in the Department plans, preference would be given to the turbine installation.

To meet this requirement the Curtis turbine advocates presented a plan having four shafts with high-pressure turbines on two inboard shafts and low-pressure turbines on the two outboard shafts. The Bureau informed them that if cruising turbines with reduction gear were fitted to the low-pressure shafts, exhausting to the main H.P. turbines, the installation would be approved. This change was made and one vessel, the *Pennsylvania,* was fitted with Curtis turbines designed for 220 revolutions per minute, and the other vessel, the *Arizona,* with Parsons turbines of 240 revolutions per minute, both of them being fitted with cruising turbines and reduction gear for economy at low speeds. These vessels were the first battleships ever laid down by any nation with this arrangement for cruising economy. Before the *Nevada* was completed the contractors, evidently fearing that the guaranteed cruising economies could not be realized, proposed to the Department that the means for obtaining this economy originally proposed be abandoned and that cruising turbines and reduction gear installed on each shaft forward of the main turbines be sub-

stituted. This change was approved and the machinery so built. The *Nevada's* turbines marked a further advance for this type in that the old form of disc diaphragms between stages was abandoned; diaphragms split on the horizontal diameter being fitted. The turbines as fitted on the *Pennsylvania* also marked a still further advance, the high-pressure turbines in this vessel being of the so-called "reverse flow" type, which means that after the steam had passed partially through the turbine, a portion of it was by-passed to the after end of the turbine and flowed forward, thus reducing the steam thrust and also, if so desired in the design, decreasing the lengths of the blades.

The economy results obtained with the three vessels which have been tried up to date, the *Nevada, Oklahoma* and *Pennsylvania,* have been very satisfactory and all expectations have been met. The low power results with the last named ship may even be classed as remarkable.

Taking the *Oklahoma's* fuel consumptions as 100 per cent., the fuel expenditures were as the following percentages:

Speeds	10	15	19	20.5
Oklahoma, per cent.	100.	100.	100.	100.
Nevada, per cent.	88.9	88.6	99.8	101.2
Pennsylvania, per cent.	91.5	103.6	101.4	101.6

This shows that the *Pennsylvania,* while over 3,000 tons heavier than the *Oklahoma* and *Nevada,* on the trials consumed very little more fuel than the reciprocating engined ship.

These results marked the passing of the reciprocating engine for battleship propulsion, and the succeeding two vessels, the *Mississippi* and *Idaho,* were practically duplications of the machinery of the *Pennsylvania* and *Arizona.*

During this same period the Department contracted for and there were constructed a destroyer tender, the *Melville,* fitted with Westinghouse reduction gear, consisting of H.P. and

L.P. turbines driving two pinion shafts with pinions meshing into a gear wheel on the single main shaft, and a submarine tender, the *Bushnell,* with Parsons reduction gear, similarly fitted. The latter vessel had, however, one innovation, and that was in the means taken to drive the generators for providing electric current to charge the submarine batteries. These generators were two in number, one located directly forward and in line with each turbine shaft to which they were attached by means of loose couplings, similar couplings also being fitted between the turbines and reduction gears. By this means both the main turbines could be used for driving the generators in port, while when under way either turbine could be used for driving the main propelling shaft while the other operated one of the generators, or both generators could be cut out and both turbines used to propel the vessel. During this same period plans were purchased for Diesel engines of 2,500 S.H.P., and work started on the construction of two of these engines for the fuel ship *Maumee.* These engines have been completed and the vessel is now in service. The *Maumee's* engines represent the maximum power installed in any one ship up to the present time. All three of the above installations have been satisfactory in operation and undoubtedly mark the passing of the reciprocating steam engine from the field of auxiliary vessels.

The forced-draft blowers on the *Oklahoma* and *Nevada* are driven by electric motors, but trials have demonstrated that there is not sufficiently close regulation of air supply for efficient oil burning with this type of drive, having its speed regulation stepped. These blowers are to be replaced with steam turbine-driven ones similar to those on the later vessels.

The next advance in propulsion of capital ships occurred when the Department decided to fit the *New Mexico* with electric propelling machinery. This decision was the result of the unqualified success of this type of machinery on the *Jupiter.* This decision has since been adhered to and there are

at present seven battleships and four battle cruisers building with such machinery.

The advantages of electric machinery for such classes of vessels are:

(1) Greatly increased torpedo protection for ships;

(2) Greater flexibility in machinery arrangement;

(3) Better and wider separation of important units;

(4) Minimum lengths and diameters of steam pipes;

(5) Fewer bulkheads pierced by steam and feed piping;

(6) Reduced heating of vessel from steam pipes;

(7) Better centralization of power;

(8) Reduced engine-room complement;

(9) Elimination of danger from fractures of piping due to shells striking protective deck;

(10) Greater ease in control;

(11) Greater flexibility in power distribution;

(12) Better maintenance of economy through a wide range of powers;

(13) No metallic contact between rotor and stator of motor;

(14) Eliminating all dangers of disarrangement due to shaft vibration, when the helm is put hard over;

(15) Maximum reduction in length of shafting;

(16) Increased backing power.

In opposition to these claims can only be advanced one, that of greater weight. This can be discounted on the heavy ships where protection is of vital importance. On the light fast vessels such as destroyers and scouts where no protection exists, everything is sacrificed for speed, and for such vessels the mechanical reduction gear stands preëminent today.

In the course of this paper I have taken you through power development from 16,500 I.H.P. to 60,000 S.H.P. for battleships, from 25,000 I.H.P. for armored cruisers to 180,000 for battle cruisers, and from 7,500 I.H.P. to 28,000 S.H.P. for destroyers. Also I have shown you the reciprocating

engine reach its maximum efficiency and disappear to be replaced by the turbine drive with and without cruising units. This again when highly successful has bowed its head and moved aside that the straight mechanical or electrical reduction gear might take its place. The end has not yet been reached, but probably changes from now on will be slow and no decidedly new type will usurp the places of those now in use for years to come. The future is in the hands of the young men of the Service, and every confidence is felt that they will not lag behind in taking their share of the burden in the development of the marine machinery of the future.

NAVY LUBRICANTS AND LUBRICATION.

By Lieutenant Commander H. T. Winston, U. S. N., Member.

(1) Lubricating Oils for Machinery.
(2) Lubricants for Cutting Tools.
(3) Greases.
(4) Graphites.
(5) Miscellaneous.
(6) Appendix: Specifications and Tests.

(1) CONTRACT LUBRICATING OILS.

General Remarks, Kinds of Oils Used, Names and Brands of Oils, Classification of Oils, Physical Properties of Straight Mineral Oils, Simple Tests for Oils, Methods of Purchasing for the Navy, Naval Contracts, Advantages of Annual Contracts.

Contracts covering over 1,000,000 gallons of lubricating oils at a total cost of more than $250,000 are being made each year by the Navy Department in order to provide the necessary lubricants for the machinery of vessels and shore stations in various parts of the world. Some other Government departments are also, by special arrangement, using quantities of these contract oils.

It will be of interest to see how these oils are selected, tested and purchased; and how certain conclusions have been arrived at, as a result of years of careful investigation and experimentation. In this work the Engineering Experiment Station at Annapolis has played a most important part, being engaged continuously in research work, chemical analyses, physical tests and other investigations.

Note.—See Appendix A for information concerning standard tests made at the Engineering Experiment Station.

See Appendix B for information concerning tests made in actual service and the use of oils on board naval vessels.

KINDS OF OILS USED.

No oils are contracted for which contain ingredients harmful to machinery or have characteristics which are likely to prevent efficient lubrication. An oil to be considered acceptable must first be proven, by actual tests, to be an efficient and satisfactory lubricant under the ordinary conditions experienced in the naval service. Results are desired and not theories or promises.

The following have been found necessary. [See (*a*), (*b*) or (*c*) or (*d*), (*e*) and (*f*).]

(*a*) *Four oils, for use in all forced-feed lubrication systems, in internal-combustion engines, for splash lubrication and for general lubrication.* These are similar straight mineral oils, but have viscosities covering a wide range (from about 140 to 400 secs., Saybolt, at 100 degrees F. for paraffine base oils, and 180 to 700 secs. for asphaltic base oils).

Probably any vessel could, by choosing two of these oils best suited to her needs, properly lubricate with only two oils everything on board needing lubrication. However, certain special oils to meet special requirements are considered desirable and are being provided. Their description follows.

(*b*) *Compounded marine-engine oil* contains a percentage of fixed oil (Navy contract oils about 20 per cent. of blown rape seed oil or something similar) which latter causes a heavy emulsion or lather to be formed upon contact with water. Such an oil is intended for use where cooling water is played on bearing surfaces, the emulsion formed sticking to the surface, instead of being rapidly washed off, and thus preventing excessive oil consumption. Manifestly this oil is highly unsatisfactory for forced-lubrication systems, since the emulsions would soon stop up some of the smaller oil pipes.

(*c*) *Oils containing only 8 or 10 per cent. of blown rape seed* or other fixed oil are not very satisfactory, especially at high speeds. The reasons for this are not entirely clear and personal equation may be an important factor. Such oils do not form a heavy lather and seem to run off bearing surfaces

easily, allowing the bearing to heat up. Possibly the mineral oil contents used in these oils have too low viscosities and do not contain the emulsifying constituents needed for sticking to the surface of the bearing. These oils also feed more rapidly than the heavier oils unless the wicks are reduced in number or the feed otherwise cut down. Hence a man familiar with the heavy oil is likely to let an oil box empty rapidly and the bearing run dry. At any rate straight mineral oils as described below have given better results in actual service.

(*d*) *Straight mineral oils, which form an emulsion with water and are exclusively a home product,* have been used with very good results in lieu of the compounded oil. Tests are continuing with this less expensive type of substitute oil. Up to the present time extensive tests on the torpedo boat *Bailey* at Annapolis, on the battleship *Nebraska* and on other vessels have shown the straight mineral oils just as satisfactory as the compounded oils. Compounded oils are likely to give trouble due to separation of the fixed oil which forms a heavy, gummy mass. The straight oil does not give a heavy lather and this may cause old engineers to condemn it at once and predict dire results as soon as a trial of the oil is started. Such fears are groundless, since the bearings do not heat up; also the machinery is cleaner and neater without the heavy lather. The engine-room force, of course, has to get accustomed to the look of bearings without the heavy lather and to the results obtained with a product quite different from the compounded oil. It will be found, for example, that excessive amounts of oil will be used unless the strands of wicking are materially reduced with wick-feed lubrication.

(*e*) *Ice-Machine Oil* is a light, straight mineral oil which must have a low cold point so it will not freeze in the cylinders of an ice machine. The lightest of the four "General use" oils above mentioned is ordinarily satisfactory for temporary use in an emergency.

(*f*) *Cylinder Oil* of high viscosity with high flash point is

NOTE.—An interesting article, by Lieutenant J. L. Kauffman, U. S. N., concerning these oils with details of tests appears in the February, 1917, number of this Journal.

used for internal lubrication of steam cylinders and air-compressor cylinders when oil lubrication is necessary. The high cold point, high internal friction, carbon deposits, and impurities often contained make these heavy, black oils undesirable for forced-feed and internal-cumbustion engine lubrication.

NAMES AND BRANDS OF OILS.

The many different names under which lubricating oils are advertised and sold tend to mystify the consumer. He is led to believe that a certain name indicates certain secret and magic lubricating qualities, contained in no other oil and which are essential to prevent the ruin and destruction of his machinery or mechanisms. It may be that the same oil, or practically identical oils, are branded with different names by an oil company, in order to meet local conditions in various places or to make it appear that each name represents just what is needed for a particular mechanism. Any other company may follow the same scheme with the result that a certain kind of oil can be purchased under a hundred different confusing names. Somewhat the same situation would exist if different producers of copper, lead and gold called their product by many different names, with variations for metals from different ores, different purities, etc. Oils might better be designated as "18 carat," "14 carat," "99 per cent. pure," or rather by equivalent simple descriptions after being separated into groupings based on actual lubrication requirements. The groupings used by the Navy Department are shown elsewhere in this article; and the suggestion, made to a number of large oil companies, that the names they give to their oils indicate important properties together with the general purpose for which intended, has met with favorable reception.

CLASSIFICATION OF OILS.

First it may be said that there are good oils and bad oils: *i. e.*, for a specific kind of lubrication it is easy to obtain almost anywhere both suitable and unsuitable oils. Also there are

many oil fields from which the crude product may be obtained and refined, producing oils of various grades of purity. The highly refined, pale, propely filtered and washed, straight mineral oils may be compared to 99½ per cent. pure copper; the unrefined oils to the unrefined ores.

Straight mineral lubricating oils are a product of the earth, obtainable by refining a good many different kinds of crudes. All these crudes contain impurities and must be treated to remove them or remove them in part.

Highly refined oils from different crudes do not have identical properties. Some have high and some low viscosities, some low and some high cold points (or pour or freezing points), some produce less carbon than others; also there are differences in flash points, fire points, evaporation under heat, gravities, etc. However, *so far as ordinary general lubrication is concerned, the differences between all these are of little importance.* All good oils may be expected to give good lubrication if objectionable impurities have been removed. Naturally, the above mentioned properties make certain oils more desirable than others for special purposes and under certain local conditions, but it should not be inferred that good lubrication can not be obtained with all. Rather the inference should be that the results are somewhat better with one or another of these oils, depending upon special local conditions. It may be mentioned that vessels have accidently gotten ordinary fuel oil mixed in with the lubrciating oil and still have been able to obtain satisfactory lubrication with this mixture.

CLASSIFICATION FOR NAVAL PURPOSES.

The following groupings have been adopted by the Navy (see also Appendix B).

(*a*) *Three or four highly refined straight mineral oils* with different viscosities for forced-feed and general lubrication. As nearly all important machinery is now equipped with some form of forced-feed lubrication where the same oil is continuously being pumped through the system, oils of this class are

used to a large extent and are of greater importance than all the others. For sea-going vessels, which are likely to get salt water into the oil used in forced-feed systems, it is extremely desirable that the oil does not contain constituents which will form permanent emulsions with sea water. In addition, the oil should readily separate from salt water after standing a short while, or in any event in a couple of hours after heating the mixture to about 200 degrees F. It is important that these oils do not form a bad emulsion with fresh water, separate easily from water, do not form heavy carbon deposits in internal-combustion engines, and contain no tallow, animal fats, paraffine wax, acids or other injurious foreign matter.

(*b*) *Compounded or imperfectly refined straight mineral oils* for use on bearings which have water played on them. The emulsion formed with water causes the oil to stick instead of being washed off. In this connection, however, it should be noted that a mixture of oil and water is not as good a lubricant as oil alone.

(*c*) *Special oils* are used for ice machines, steam cylinders, and mechanisms of torpedoes. These are required to be a straight mineral product.

PHYSICAL PROPERTIES OF STRAIGHT MINERAL OILS.

The important lubricating properties are viscosity, cold (or pour) point, emulsibility with water (cold and hot), and carbon *actually deposited* in an internal-combustion engine. Carbon passing through the engine is of little importance.

Unimportant properties are gravity, fire point, flash point and frictional resistance. Of these gravity simply helps to identify oils from different fields. The others do not vary enough in good oils to be of importance except in special cases. The flash and fire points vary considerably, but none are low enough to be dangerous, and none are high enough to prevent complete combustion when an explosion takes place in an internal-combustion engine. Frictional resistance is always small

and does not vary greatly, but it should be noted that high viscosity produces the highest internal friction.

Properly refined oils are clear and pale. "Red" oils are not the highest grade oils and contain certain ingredients which have not been removed. They are cheaper than the others, but may be entirely suitable for use where emulsions or possible carbon deposits are immaterial.

Mixing similar good oils does no harm, and, contrary to belief in some quarters, the resultant product shows no unexpected nor abnormal properties. The lubricating and physical properties of the mixture are simply, for all practical purposes, a mean of those of the original oils. For example, a gallon of oil with 150 secs. viscosity and a 370-degree flash point is mixed with two gallons of oil which has 300 secs. viscosity and 400-degree flash point. It will be found that the mixture has about 250 secs. viscosity and 390 degrees flash point.

"Worn out" oils, oils "with the body gone," with "no life in them," etc., may be frequently heard mentioned in a very indefinite sort of way by those concerned with lubrication. This is most often the case where the same oil is pumped again and again through a forced-feed system or where splash lubrication is used for gasoline engines, as in automobiles, for example.

In some cases it is recommended by engine manufacturers that all the "old oil" be thrown away and replaced at short intervals with "fresh," new oil. Naturally, this greatly increases the amount of oil used; also the quantity purchased from the "oilman"!!

There is little evidence to show that a *good* oil "wears out" or loses any essential lubricating properties under ordinary circumstances. High heat will, of course, cause some "light" oils to become "heavier" oils, due to the distillation or evaporation of a percentage of the original product; but this does not change a good oil into a bad oil. The result is simply a somewhat reduced amount of good oil with an increased viscosity.

Certain oils may produce heavy carbon deposits, contain in-

gredients which in time produce acids, form bad emulsions with water, etc., as previously explained. These, which have been classed as bad oils, together with dirt, rust, metal filings or other foreign matter, which can generally be kept out of the oil, most frequently account for the "worn out" oils. With gasoline engines, when there is a leakage of gasoline into the oil, naturally there is produced in a short time a new mixture which is a poor lubricant.

The engine manufacturer desires to insure good lubrication of his machine, maybe the oil producer wants to sell more oil. Both can chime in unison "Use plenty of oil and renew it frequently."

Apparently the lubricating-oil producer who supplies a good oil at a reasonable profit and shows how it can be used economically, should be able to do a large business and be ready for severe competition whenever the trade is slack.

Interesting tests have been made on "used" lubricating oils, and a full description of such tests will, no doubt, appear in this Journal.

Good oils used in forced-feed lubrication system show after about 1,000 hours' use some oxidation, but the lubricating properties were not materially changed. The cold points were raised 10 degrees to 20 degrees F., there was an increase in the formation of organic acids and of resins. These ingredients might, after long use, increase to percentages seriously affecting lubricating properties. However, such harmful ingredients are easily removed by filtering the oil through animal charcoal, the filtered product being as good or possibly better than the original oil.

SIMPLE TESTS FOR OILS.

A few simple tests may be made by almost any one and will be of assistance in rejecting unsuitable oils. These are, of course, only crude methods not comparable with thorough and complete tests.

A good oil, before use, will appear a pale yellow color, will not separate into layers of different shades or deposit sediment

after standing a few days; it will not show rusting or corrosion of pieces of polished steel which have been suspended in it. When mixed with warm water and shaken in a bottle the oil will soon separate from the water and show no soapy emulsion. (Marine engine and black cylinder stock oils will show this emulsion.) When tested with litmus paper, there will appear no acid contents, and when cooled with ice the oil will not show a cloudy appearance indicating waxy contents.

METHODS OF PURCHASING FOR THE NAVY.

After advertisement in the open market, blank proposals covering a year's lubrication requirements are distributed to prospective bidders, and the bids (with bonds or other guaranty attached) are received and opened in the Bureau of Supplies and Accounts on a fixed date. These bids are then referred to the technical officers who are concerned with lubrication for recommendation as to award. All oils not thoroughly tried, tested and known to be satisfactory are excluded; consideration is given to low bids for groups of items and the ability of the bidders to make the required deliveries promptly, and awards are made to those considered the lowest acceptable bidders.

NAVAL CONTRACTS.

The contracts require deliveries of the various oils, as needed during the current year, to the various ships and naval stations, which are separated into groups—Atlantic, Pacific, Oriental, and Miscellaneous Foreign.

Generally one bidder receives the award for all the important oils in one group, which is an advantageous arrangement since only one set of agents is, in this way, needed at the delivery points, and supply officers have to deal only with one contractor.

All oils are required guaranteed the same as the samples previously tested and found satisfactory. If they are not as represented, other oils may be purchased in the open market and

charged to the contractor's account, thus causing him both a monetary loss and a loss in prestige.

Careless contractors or jobbers, who ship some other oil than the one called for or different kinds of oil under the same brand name, are a nuisance, to be avoided if practicable. If a firm deliberately sends in a sample of one oil for test and then supplies other kinds under contract, little consideration is due such a concern. The Navy frequently makes check tests of oils actually received under contract.

ADVANTAGES OF ANNUAL CONTRACTS.

Large consolidated purchases not only materially reduce costs, but also are of very considerable importance otherwise in the case of oils. Oils can not well be covered by definite specifications; complete tests require much time and labor; sales agents make extravagant claims which are not easily refuted by the uninitiated; local prejudice or influence may have to be combatted; and much work is involved by each contract in connection with proposals, examination of samples, consideration of bids, etc. With an annual contract all that is necessary is for the supply officer, or engineer, to order the kind of oil needed from the contractor, who will supply a product known to be satisfactory.

(2) LUBRICANTS FOR CUTTING TOOLS.

Machine-cutting-tool lubricants appear on the market in both the paste and liquid forms and in almost endless variety. They may consist largely of cheap soft soap or soap and water, or contain acids, mal-odorous fats or oils, strong alkalis, gumming constituents and other objectionable ingredients. Local influence, and prejudice or ignorance, at times play an important part in bringing about the purchase at a high price of a cheap product.

Naturally, if mostly soft soap or 50 per cent. water is desired, instead of high-grade oil, such a compound should be specified and paid for at the current market rates.

Consolidated annual contracts provide for naval shore stations three lubricants, which must be in accordance with the specifications shown in Appendix " C."

Brands which have successfully passed the tests are added to acceptable lists and require no further tests. It will be noted that a special improvised apparatus is used to determine " cooling efficiency," which is considered important.

In general, it may be assumed that when a product is claimed to be about three times as efficient as anything else on the market, others selling at a lower price are better. To be able to sell a cheap product at a high price extravagant claims, extensive advertising, etc., have to be resorted to. The consumer pays for all these and in addition is assessed for the heavy profit.

(3) LUBRICATING GREASES.

Hard and medium mineral lubricating greases (for use in grease cups) and graphite lubricating grease (for use on gears or elsewhere when desirable) are largely used in the Navy. They are very well described in the specifications shown in Appendix D. The ingredients consist mostly (80 to 90 per cent.) of mineral oils, which have been heretofore discussed, and it will be noted that a practical test to insure certain lubricating and cooling properties is provided for. It is considered that the elimination of the small percentage of animal fats called for by present specifications is desirable. These might, under certain conditions, become rancid and form acid compounds.

(4) GRAPHITES.

High-grade flake graphite or electrically-prepared graphite, containing at least 88 per cent. of graphitic carbon and, of course, free from dirt, is used in connection with oils and greases to give them more body or consistency and prevent rapid melting or lack of adhesion under high temperatures. These graphites are particularly valuable in smoothing up a rough bearing surface, filling up small crevices and hollows and producing a glazed surface with consequent reduction of

friction. When bearing surfaces are smooth the use of graphite does not appear beneficial. In fact, its excessive use is likely soon to clog oil holes or piping, thus preventing the proper lubrication of the bearings. For good lubrication it is necessary to provide a flow of oil through the bearing in order to provide a low coefficient of friction, wash away dirt particles and carry off heat.

It is considered objectionable to use much graphite in any place where it will go through surface condensers, boiler feed piping, etc., finally getting into the boilers. Experiments indicate that graphite in boilers tends to aid pitting or corrosion there. Great care is taken to prevent anything except Navy standard boiler compound getting into boiler feed water.

Ordinary ground amorphous graphite is used only as a paint ingredient, and not on bearing surfaces.

(5) MISCELLANEOUS.

Special lubricants are used for the mechanisms of torpedoes and high-pressure air compressors. Only a few special oils which have been found satisfactory in service are purchased under the supervision of the Bureau of Ordnance.

Standard specifications cover the following special materials: lard oil, sperm oil, mineral sperm oil, glycerin, neatsfoot oil and soap. The last is occasionally used in connection with cooling hot piston rods. Flour of sulphur is used at times to help cool hot bearings.

Vaseline or *Petrolatum* is required to be of good quality, clean, refined and free from acidity or adulteration. White petrolatum is not accepted.

Tallow must be pure and refined, free from rancidity, dirt, cracklings, soap, or other similar ingredients, and free from more acidity than the equivalent of 2 per cent. of oleic acid.

Tallow mixed with white lead in oil (pure hydrated carbonate of lead, free from adulterants) is used to slush idle mechanisms and protect them from the weather or atmosphere. Also special slushing compounds are at times bought for this purpose.

APPENDIX A.

INFORMATION CONCERNING TESTS OF LUBRICATING OILS AT THE ENGINEERING EXPERIMENT STATION.

ISSUED BY THE BUREAU OF STEAM ENGINEERING.

SUBJECT TO CHANGE OR MODIFICATION AS FOUND DESIRABLE.

February 7, 1916.

1. All oils not known to be satisfactory are subject to test at the Engineering Experiment Station, Annapolis, Md., and to a further test in service on board ship when considered necessary, before being considered acceptable by the Bureau of Steam Engineering.

2. Authorization of an oil test must be obtained from the Bureau of Steam Engineering, which Bureau directs the Engineering Experiment Station to conduct the test. The authorized tests are then given a number and are taken up in regular order; persons interested being permitted to visit the Experiment Station and inspect the methods of testing employed.

3. All expenses of tests are borne by the concern requesting the same; the necessary quantities of oils being shipped to the Supply Officer, Naval Academy, Annapolis, Md., *marked "For Test at Engineering Experiment Station,"* and with all express or other charges prepaid. Before the test is started a check made payable to the *Superintendent, U. S. Naval Academy,* must be mailed to that official at Annapolis, Md., to cover the costs of tests; any unexpended balance being returned to the exhibitor.

4. All tests are made with the strict understanding that they are for the information of the Government only and that results of tests are not to be used as an advertisement.

5. The cost of tests varies with the labor involved and the number of oils tested, the expense being reduced where an oil proves unacceptable in the early stages of the test.

6. In general a deposit of $50.00 is required for a test of one oil and $200.00 for 6 or 8 oils, the amounts of each oil required by the Experiment Station being from 5 to 20 gallons.

7. The complete test of an oil consists of three parts, namely: chemical, physical and practical tests.

8. To successfully pass the *chemical* tests *all* oils should be neutral in reaction and should not show the presence of moisture, matter insoluble in petroleum ether (hard asphalt), matter insoluble in ether alcohol (soft asphalt), sulphur, charring or waxlike constituents, naphthenic acids, sulphonated oils, soap, resin, or carry constituents, the presence of which indicates adulteration or lack of proper refining. Except in oil for engines without forced lubrication, no traces of fixed oils (animal or vegetable fats) should be found.

9. In lubricating oil for main engines without forced lubrication, ap-

proved fixed oils, such as rape seed, olive, tallow, lard and neatsfoot oil may be used. When the above fixed oils are used, they will be well refined with alkalies, unadulterated, containing a minimum of free fatty acids, with no moisture or gumming constituents. Olive oil should not have a high specific gravity. If satisfactory emulsifying results can be obtained with straight mineral oils on engines without forced lubrication they may be submitted for service test.

10. The *physical* tests applied to *each* oil are as follows:

(a) Specific gravity—pyknometer.
Baumé gravity.

(b) *Flash* and *Fire Point* (Cleveland open cup, Pensky Martin closed cup).
The flash point should not be below 300 fahr., closed cup; and for steam cylinder oil not below 450 fahr.

(c) *Cold Point.* The cold point should not be above 32 fahr., except for cylinder oils. For ice machine oils it should be as low as possible, at least low enough for the operating condition of minus 15 fahr. in a dense air ice machine.

(d) *Viscosity.* The Saybolt Universal Viscosimeter is used. The viscosity of the oils must be sufficient for the purpose intended, and, except for ice machine oils, must not be less than 140 seconds at 100 fahr. Viscosity is taken at 100, 130 and 210 fahr.

(e) *Emulsion tests.* Emulsion tests are made on all straight mineral oils, except cylinder oils. Four emulsion runs are made, using 40 c. c. of oil in each case and
(1) 40 c. c. of distilled water;
(2) 40 c. c. of salt water;
(3) 40 c. c. of normal caustic soda;
(4) 40 c. c. of boiling distilled water.
The mixture is stirred with a paddle for five minutes at 1500 revolutions per minute, the mixture being kept at a temperature of 130 fahr. during the stirring and while settling out. On oils used with forced lubrication or on ice machines the oil must completely settle out in less than 30 minutes.

(f) *Friction tests* are made on the Experiment Station friction machine to determine the relative friction of various oils under varying conditions of speed, bearing-cap pressures, and temperatures.

11. *Practical tests* of oils for forced lubrication are made on the Engineering Experiment Station's 75-kw. turbo-generator, making 2,400 revolutions per minute, with the temperature of the bearing varying from 180 to 210 fahr. Tests of ice-machine oils are made on the ½-ton Allen Dense-Air Ice Machine. These tests cover a working period of at least 100 hours.

12. Upon the completion of the tests at the Engineering Experiment Station a service test is usually recommended on all but cylinder oils. These

tests are made on ships in commission, and cover a period of several months. The oil for this test is paid for at the price of the contract oil in use for the purposes intended.

13. Oils should be submitted as follows:

(a) Three oils (light, medium and heavy) to be suitable for turbines, all forms of forced-feed or splash lubrication, dynamo engines, and all internal-combustion engines (including Diesel type). These three oils should cover the required range of viscosity. The pour point must not be above 15 degrees F. for the medium and light oils.

(b) Marine engine oil suitable for main engines without forced lubrication. This oil should form an emulsion when mixed with water, and may be either a compounded oil or a straight mineral oil. This oil must feed efficiently with wick feed, and not be washed off when mixed with water.

(c) An oil for use on ice machines in both the compressor and expander cylinders. This oil should not give off an objectionable odor when used with the Dense-Air Ice Machine.

(d) Oil for steam-cylinder lubrication. This oil must be a straight mineral oil.

(e) Oil for steam cylinders using superheated steam. This oil will be used only on shore stations and should have a high flash point.

14. It is desired to have the trade names of the oils indicate the use for which they are submitted. For instance, "Cetus 200 Mineral" would indicate, first, the trade name: "*Cetus*"; second, the viscosity, Saybolt at 100 fahr.: "200"; and, third, the kind of oil: "Mineral". For marine engines without forced lubrication a name such as "Neptune Marine Engine" or "Atlantic Marine Engine" would indicate the use, no matter whether the oil were a compounded or a straight mineral oil. For ice-machine oils names such as "Polar *Ice* Machine" or "Spica *Ice* Machine" will do. And for cylinder oils such names as "Pinnacle *Mineral Cylinder*" or "600-W *Mineral Cylinder*" are suitable.

15. It is necessary that the trade name of the oil furnished the Engineering Experiment Station for the original test be the same as that furnished for service tests.

16. When the contract is awarded, the oils furnished on contract must be identical with those sent to the Engineering Experiment Station for the original acceptance test.

17. As correspondence and arrangements in connection with tests, the actual conduct of tests, and unforseen contingencies, frequently consume many months, *prospective bidders should submit their oils* for test *eight months before they intend to bid* for Government contracts.

18. *Oils for Torpedo Lubrication* and *Torpedo Air Compressors* are under the *Bureau of Ordnance* and subject to the tests prescribed by that Bureau.

APPENDIX B

NAVY DEPARTMENT.
BUREAU OF STEAM ENGINEERING.
November 20, 1916.

LUBRICATING OILS.

SERVICE TESTS AND GENERAL INSTRUCTIONS.

1. In order that satisfactory lubricating oils may be supplied under annual contract to the whole naval service it is necessary that certain types of oils be given actual tests in service to determine whether or not they should be contracted for

2. The Bureau's policy amounts to the establishing of acceptable lists of lubricating oils and restricting contracts or purchases to such oils. All oils sent in for approval are passed upon by the Engineering Experiment Station and those which would evidently be injurious or unsatisfactory are rejected. However in the case of certain oils service tests only will demonstrate whether or not they will give entirely satisfactory results. When service tests of oils are directed it may be assumed that such oils will at least give fair results and will not prove injurious to machinery.

GENERAL LUBRICATION.

3. There are needed 3 or 4 *oils for general lubrication.* These are really one type of oil, graded according to viscosities; and ranging from the high-viscosity, thick oil to the low-viscosity, thin, freely flowing oil. For proper identification the contract oils are being called "Extra heavy," "heavy," "medium," and "light" oils (arbitrary grading of the Bureau of Steam Engineering). Such oil must not be injurious to machinery, must separate easily when mixed with water and must not form a heavy emulsion which will clog oil pipes. In general the heavy oils are required where the temperature is high, for heavy, slow-moving machinery and where there would be excessive leakage of light oils in the lubricating system. Light oils are needed in cold climates, when a rapid flow of the oil is required for cooling purposes, and for small rapidly-moving pieces of machinery.

KINDS OF OIL NEEDED.

4. The above "*General Use*" oils are intended for forced-feed and splash lubrication (turbines, gas engines, Diesel engines, aeroplane engines, reciprocating engines, etc.) and general use, except as below:

Special Oils are needed for.

(a) *Wick or sight-feed* lubrication systems, *where water playing on the bearing surfaces* would quickly wash away the oil and cause excessive oil consumption. In such cases a compounded "Marine engine oil" is used. It contains 10 per cent to 20 per cent of a special oil, which forms a heavy emulsion (or lather) with water, and sticks to the bearing surface. *This oil would quickly cause trouble and plug up the piping of a forced-feed lubrication system.*

(b) *Ice-machine expander and compressor cylinders.* The oil must be free from odors and remain liquid at very low temperatures.

(c) *Steam Cylinders.* Must be a heavy oil which will stick to the cylinder walls, with a high flash point so it will more effectively resist heat.

(d) *Hot-Running Torpedoes* and *Torpedo Air Compressors.* Two oils provided under the Bureau of Ordnance to meet special requirements.

5. The *objects of service tests of oils* will be to determine their suitability for general service use under various conditions, including high speeds. General efficiency and oil consumption as compared with contract oils should be ascertained. Chemical and laboratory tests need not be made by a vessel, unless there are very good reasons to believe the oil is not the same as that tested at the Engineering Experiment Station. Not less than a gallon sample of oil is necessary for casual examination and test at the Engineering Experiment Station.

6. The *amount of oil needed for test* will be obtained on requisition, from the designated oil company, by the vessel. Such requisition should state "This proprietary oil is required for special test as directed in Bureau of Steam Engineering letter No. .... (date.....)." The cost of the oil delivered must not exceed the cost of similar contract oil.

7. The *report of the service test* should cover not less than 10 days' steaming and, as far as possible, show the following:

Amount of test oil used.

Number of hours oil was used.

Average speeds of machinery.

Maximum and average temperature of bearings.

Maximum and average temperature of oil in system.

Average pressure carried on oil pumps.

Gallons of oil per hour ordinarily pumped through system.

Temperature at which oil is boiled out and amount of water drawn off.

Oil consumption as compared with the ordinary oil.

Unusual conditions or attendant circumstances.

NOTE.—On twin-screw vessels the test oil is generally tried on one engine and contract oil used on the other. When a test oil is considered unsatisfactory 5-gallon samples of both the used and unused oil may be required sent to the Engineering Experiment Station for further test and report.

APPENDIX C.

NAVY DEPARTMENT SPECIFICATIONS, 14-C-3b.

January 2, 1917.

CUTTING COMPOUND, (PASTE).

GENERAL SPECIFICATIONS—1. General Specifications for the Inspection of Material, issued by the Navy Department, in effect at date of opening of bids, shall form part of these specifications.

PURPOSE—2. To be used for machine cutting tool lubricant when mixed as directed.

COMPOSITION—3. To be a soluble paste compound consisting of an alkali soap, mineral and fixed saponifiable oils and water. To be free from disagreeable odors, mineral acids, or any ingredients injurious to persons handling the material.

To contain not more than 25 per cent. of water, not more than 20 per cent. of alkali soap, not less than 40 per cent. of mineral oil, and the remainder fixed saponifiable oil. It must form a stable emulsion when mixed with water.

LUBRICATION—4. The emulsion must sufficiently lubricate turret and automatic machines to prevent sticking, the solution used to be suitable for work being performed on the machine, and must show no tendency to leave a gummy residue.

CORROSION—5. Strips of polished steel are to show no appreciable corrosion after being partly immersed in mixture for a period of two weeks.

PHYSICAL TEST—6. One pound of the paste will be put into emulsification with 3 gallons of water and the emulsion permitted to flow at the rate of 1 gallon per minute over a steel cylinder heated by an electrical coil consuming 440 watts which maintains a constant temperature of 100 degrees C. in air. After a period of 8 hours, the maximum rise of temperature of the emulsion shall not exceed 12 degrees C. This physical test will be conducted at the New York Navy Yard on samples before approval on the standard apparatus shown in drawing No. 36367-A, which may be obtained from the Engineer Officer at the navy yard, New York.

DELIVERY—7. To be purchased by the pound and delivered in heavy barrels of not more than 500 pounds capacity, suitable for foreign shipment, or in 25-pound friction-top cans packed in wood cases, two to a case. The quality of the containers to be such as to provide against leakage or deterioration in storage and in handling.

The barrels to be plainly marked with trade-mark of material, name of manufacturer, contract number, quantity, and instruction for use of compound.

Prices are net on compound delivered in barrels. Barrels to remain the property of the contractor.

All barrels in good condition will be returned by the Government at the contractor's expense within six months after date of each delivery of the compound to shore stations. All barrels not returned within this period will be paid for at the rate of $1.00 per barrel, chargeable to the appropriation under which the material was purchased or used.

ACCEPTABLE LIST OF CUTTING COMPOUND (PASTE)—8. Contractors who propose to furnish cutting compound (paste) under these specifications may have a 5-pound sample tested at navy yard, New York, and if the test proves satisfactory this cutting compound (paste) will be placed on an acceptable list. The contractor will be required to submit a 5-pound sample of cutting compound (paste) to be supplied with each new bid, and state the approximate date of test.

RESTRICTIONS—9. Mineral lard oil, and soluble cutting oils will not be considered in connection with bids for cutting compounds (paste form), but will be bought under separate specifications.

NAVY DEPARTMENT SPECIFICATIONS, 14-O-10a.

March 1, 1916.

SOLUBLE CUTTING OILS OR CUTTING COMPOUNDS (LIQUID FORM).

GENERAL SPECIFICATIONS—1. General Specifications for the Inspection of Material, issued by the Navy Department, in effect at date of opening of bids, shall form part of these specifications.

PURPOSE—2. To be used in emulsion with water for machine cutting-tool lubricant.

COMPOSITION—3. To be a clean and homogeneous mixture of soluble alkali soap in mineral and fixed saponifiable oils. It shall be free from disagreeable odors, sediment, mineral acids, ingredients injurious to persons handling, and shall contain not more than 10 per cent. of water and not more than 20 per cent. of soluble alkali soap.

EMULSIFICATION—4. To be capable of readily mixing with water in all proportions, without the use of sodium carbonate or other addition, to form a stable emulsion.

LUBRICATION—5. The emulsified oil must sufficiently lubricate turret and automatic machines to prevent sticking, the solution used to be suitable for the work being performed on the machine, and must show no tendency to leave a gummy residue.

CORROSION—6. Strips of polished steel are to show no appreciable corrosion after being partly immersed in the emulsion for a period of two weeks.

PHYSICAL OR COOLING EFFICIENCY TEST—7. When 3 pints of oil are put into emulsification with 3 gallons of water and permitted to flow at the rate of 1 gallon per minute over a steel cylinder heated by an electric coil consuming 440 watts designed to maintain a constant temperature of 100 degrees C. in air for a period of eight hours, the maximum rise of temperature of the emulsion shall not exceed 12 degrees C. This physical test will be conducted on samples on a standard apparatus at the Navy Yard, New York, constructed in accordance with drawing 36367-A which may be obtained from the engineer officer of the New York Navy Yard.

DELIVERY—8. To be purchased by the gallon and delivered in heavy barrels, suitable for foreign shipment, of about 50 gallons capacity, or 5 gallon shipping cans, as required. The quality of the containers and packing cases to be such as to provide against leakage or deterioration in storage and in handling. Containers to be plainly marked with trade-mark of material, contract number, manufacturer, and quantity contained in gallons.

Prices are net on oil delivered in barrels. Barrels to remain the property of the contractor.

All barrels in good condition will be returned by the Government at the contractor's expense within six months after date of each delivery of the oil to shore stations. All barrels not returned within this period will be paid for at the rate of $1 per barrel, chargeable to the appropriation under which the material was purchased or used.

ACCEPTABLE LIST OF SOLUBLE OILS—9. Contractors who propose to furnish soluble oil under these specifications may have a 1-gallon sample tested at the New York Navy Yard, and if the test proves satisfactory, these oils will be placed on an acceptable list. The contractor will be required to submit a 1-gallon sample of the oil to be supplied with each new bid and state the approximate date of test.

RESTRICTIONS—10. Mineral lard oil and cutting compounds (paste form) will not be considered in connection with bids for soluble cutting oils, but will be bought under separate specifications.

NAVY DEPARTMENT SPECIFICATIONS, 14-O-9ao.

February 1, 1916.

OIL, LARD, MINERAL.

GENERAL SPECIFICATIONS—1. General Specifications for the Inspection of Material, issued by the Navy Department, in effect at date of opening of bids, shall form part of these specifications.

PURPOSE—2. To be used for machine cutting tool lubricant, either unadulterated or compounded with mineral oil or soda and water.

COMPOSITION—3. To be clean and homogeneous; free of disagreeable odors, rancidness, sediment, or ingredients injurious to persons handling the material; and to be easily soluble and retain oily consistency in kerosene or soda and cold-water mixtures. To have a flash point in an open tester of not less than 180 degrees C., and flow at —4 degrees C. To contain not less than 25 per cent. and not more than 35 per cent. of fixed saponifiable oils, from 60 to 70 per cent. of mineral, and not more than 5 per cent. of free fatty acid (calculated as oleic acid).

VISCOSITY—4. Measured in a Saybolt viscosimeter (with a 30 seconds water rate at 15 degrees C.) the oil to show about 185 seconds at 38 degrees C. and 115 seconds at 48 degrees C.

GUMMING—5. A saucer with enough test oil to cover the bottom when placed in an oven at a constant temperature of 120 degrees C. for a period of 8 hours, when taken out and permitted to cool gradually, shall show no signs of a gummy residue.

CORROSION—6. Strips of polished steel to show no appreciable corrosion in two weeks' time when partly immersed in samples of the oil, or in a mixture of the oil and kerosene, or in an emulsion of the oil, soda, and water.

PHYSICAL TEST—7. Three gallons of the oil unadulterated will be put into a steel tank and pumped at the rate of 1 gallon per minute over a steel cylinder heated by an electric coil consuming 440 watts which maintains a constant temperature of 100 degrees C. in air. After a period of 3 hours the maximum rise of temperature of the oil shall not exceed 30 degrees C. This physical test will be conducted at the navy yard, New York, on all samples before approval on the standard test apparatus shown in drawing 36367-A, which may be obtained from the engineer officer, navy yard, New York.

DELIVERY—8. To be purchased by the gallon and delivered in heavy bar-

rels, suitable for foreign shipment, of about 50 gallons capacity, or 5-gallon shipping cans, as required, the quality of the containers and packing cases to be such as to provide against leakage or deterioration in storage and in handling. Containers to be plainly marked with trade-mark of material, contract number, manufacturer, and quantity contained in gallons.

Prices are net on oil delivered in barrels. Barrels to remain the property of the contractor.

All barrels in good condition will be returned by the Government at the contractor's expense within six months after date of each delivery of the oil to shore stations. All barrels not returned within this period will be paid for at the rate of $1 per barrel, chargeable to the appropriation under which the material was purchased or used.

ACCEPTABLE LIST OF OILS—9. Contractors who propose to furnish mineral lard oil under these specifications may have a 5-gallon sample tested at navy yard, New York, and if the test proves satisfactory these oils will be placed on an acceptable list. The contractor will be required to submit a 1-gallon sample of the oil to be supplied with each new bid and state the approximate date of test.

RESTRICTIONS—10. Cutting compounds (paste), and soluble cutting oils, will not be considered in connection with bids for mineral lard oil, but will be bought under separate specifications.

APPENDIX D.

NAVY DEPARTMENT SPECIFICATIONS, 14-G-1b.

November 1, 1916.

MINERAL LUBRICATING GREASE.

GENERAL SPECIFICATIONS—1. "General Specifications for Inspection of Material," issued by the Navy Department, in effect at date of opening of bids, shall form part of these specifications.

COMPOSITION—2. Mineral lubricating grease shall be a homogeneous mixture consisting exclusively of from 80 to 90 per cent. of mineral oil and the remainder an odorless lime soap made from clear animal fats and the proper amount of lime for saponification. It shall be free from fillers, uncombined lime, gritty substances, rosin oil, rosin, or resinates, and from mineral or fatty acids, alkalies, or any deleterious impurities, and shall not yield more than 2 per cent. of ash from *medium grease*, and not more than 3 per cent. of ash from *hard grease*. The grease shall lose not more than 2 per cent. of its weight when heated for one hour at 110 degrees C. in a glass crystallizing dish containing about 10 grams of grease, in an air oven.

GRADES—3. (*a*) *Medium grease* shall flow at a temperature of from 75 degrees to 80 degrees C. when tested in a glass crystallizing dish containing about 10 grams of grease, heated in an air oven.

(*b*) *Hard grease* shall flow at a temperature of about 90 degrees C. when tested in a glass crystallizing dish containing about 10 grams of grease, heated in an air oven.

LUBRICATING PROPERTIES—4. The grease shall possess lubricating properties determined by practical test in a lubricant-testing machine, as follows:

When fed at a rate of not exceeding 1-½ grains per minute through a grease cup, on friction surface of a brass shoe having 9 square inches bearing surface, sustaining a load of 1,926 pounds against a steel journal 6 inches in diameter, revolving at a surface velocity of 405 feet per minute, it shall maintain an even temperature of not more than 50 degrees C. above surrounding normal temperatures, and the coefficient of friction shall be constant during the last hour of the run and shall not exceed 0.013.

PURPOSE—5. Mineral lubricating grease is intended for use in compression grease cups for bearings. For rapid-running machines in cool climates *medium grease* should be ordered. For hot climates or heavy-running machinery *hard grease* should in general be ordered.

DELIVERY—6. The grease shall be delivered in friction-top cans of 10 pounds' capacity, properly labeled with name and grade of material, manufacturer's name, and net contents of can. To be packed in boxes of 80 or 100 pounds each; the boxes to be made of ⅞ inch new pine or spruce, planed on both sides, and properly labeled with contents and contract number.

NAVY DEPARTMENT SPECIFICATIONS, 14-G-2b.

November 1, 1916.

GRAPHITE LUBRICATING GREASE.

GENERAL SPECIFICATIONS—1. General Specifications for Inspection of Material, issued by the Navy Department, in effect at date of opening of bids, shall form part of these specifications.

MATERIAL—2. Graphite grease to consist of 8 to 10 per cent. of flake graphite containing at least 82 per cent. of graphitic carbon mixed with a mineral lubricating grease of the following composition and consistency: Mineral grease to be a homogeneous mixture consisting of from 80 to 90 per cent. mineral oil, and the remainder an odorless lime soap made from clean animal fats and the proper amount of lime for saponification. To be free from grit, rosin, or resinates, and from mineral or fatty acids, alkalies, or any deleterious impurities. Medium grease to flow at a temperature of from 75 degrees to 80 degrees centigrade, and hard grease to flow at about 90 degrees centigrade when tested in a glass crystallizing dish containing about 10 grams of grease, heated in an air oven. The grease to lose not more than 2 per cent. of its weight when heated for one hour at 110 degrees centigrade and tested in a similar dish and oven.

LUBRICATING PROPERTIES—3. To possess lubricating properties, determined by practical test in Riehle bearing testing machine, as follows:

When fed at a rate not exceeding 2½ grains per minute through grease cup, on friction surface of a brass shoe bearing having 9 square inches bearing surface, sustaining a load of 1.926 pounds against a steel journal

6 inches in diameter, revolving at a surface velocity of 405 feet per minute, it shall maintain an even temperature of not more than 50 degrees centigrade above surrounding temperatures, and the coefficient of friction shall be constant during the last hour of the run and shall not exceed 0.031 for medium grease and 0.04 for hard grease.

PURPOSE—4. Graphite lubricating grease under these specifications is intended for use on gearing of heavy machinery and bearings exposed to weather and heat. If harder grease is required for special purposes, a larger percentage of graphite may be specified or graphite will be added to the grease supplied.

DELIVERY—5. To be delivered in friction top cans of 10 pounds' capacity, properly labeled with name of material, manufacturer's name, and net contents of can. To be packed in boxes of 100 pounds each; the boxes to be made of ⅞-inch new pine or spruce, planed on both sides, and properly labeled with contents and contract number.

INTEGRATING TACHOMETERS.

By COMMANDER L. A. KAISER, U. S. N., MEMBER.

Naval vessels, from the nature of their service, operate together in formation underway. To attain and maintain correct position in formation the necessity arises of providing means of controlling the speed of the engines, and this, in turn, presumes means of *measuring* or *regulating* speed. *Regulating* devices for main engines at the present time are neither practical nor desirable, but it is possible they may be built, in the future development of the electric drive. The different methods of revolution measurement fall into two general classes: first, *differential* speed indicators, which show the speed at any instant, or at successive instants; second, *integrating* speed indicators, which, by summing up the movements of the shaft, show the amount lost or gained in an interval of time.

The numerous types of speedometers — centrifugal, electric, magnetic, etc.—are examples of the first class; also the first and most primitive method of measuring revolutions (simultaneous reading of the engine-room counter and clock). Recently highly developed forms of the latter type have been brought out in which the "counter and clock" method is hardly recognized, but on examination of the underlying principles the identity is perceived. In slow-moving engines it was probably exact enough for the man on watch at the throttle to *count* the number of revolutions (or watch the counter) during a prescribed interval of time, say one minute or five minutes; in later types this is accomplished *mechanically*. It is, of course, necessary to watch the indicator con-

stantly or else assume that the speed when unwatched is the same as that at the instant of observation.

In the *second* class, the *integrating* speed indicators, variable gearing as a rule is interposed between the shaft and a pointer which revolves concentrically with a pointer revolved by clock work (the second hand of a clock); the speed indicator being set for the desired number of revolutions, the engine or shaft pointer is kept in line with the clock pointer, in the same way that the port and starboard engines are kept together by the revolution pointers or tell-tales (usually mounted on center-line bulkheads). There is no necessity for *constant* observation, as the movement of the engine pointer is the integration of the motion of the shaft, and a gain or loss is manifested by the separation of the hands as in the engine tell-tales or revolution indicators; by operation of the throttle, bringing the engine hand coincident or in line with the clock hand, the average revolutions prescribed are maintained. An integrating speed indicator will therefore show at any time that the prescribed speed has been maintained, or that the engines have been *gaining* or *losing* and, further, the *amount* of gain or loss.

A differential speed indicator shows the speed at the *moment of observation only.*

There is also this distinction in the *functions* or *capabilities* of the integrating speed indicators, and as it has an important bearing on the tactical question of maintaining position in formation it will be explained with some detail.

With an integrating speed indicator it will be shown it is possible for the *engine room to gain, or lose, distance as directed from the bridge.* The bridge determines by stadimeter, or otherwise, that the ship is *ahead* or *astern* of position, and directs the engine room to *lose* or *gain.* By consulting a previously prepared table kept near the throttle the man on watch sets the *engine hand ahead* or in the *rear* of the clock, and then by the throttle causes the engine hand to *lose* or *gain* until it is in coincidence.

If the variable gearing is so designed that the engine hand

makes *one revolution per minute* when the engine is making the prescribed speed, then it is evident that it is simply necessary to keep the engine hand coincident with the second hand of a clock (which also makes one revolution per minute).

To gain or lose distance, then, set the *engine hand* a certain number of seconds in *rear* or *advance* of the second hand and bring in coincidence with the throttle. The number of seconds to gain (or lose) ten yards is determined by the following formula:

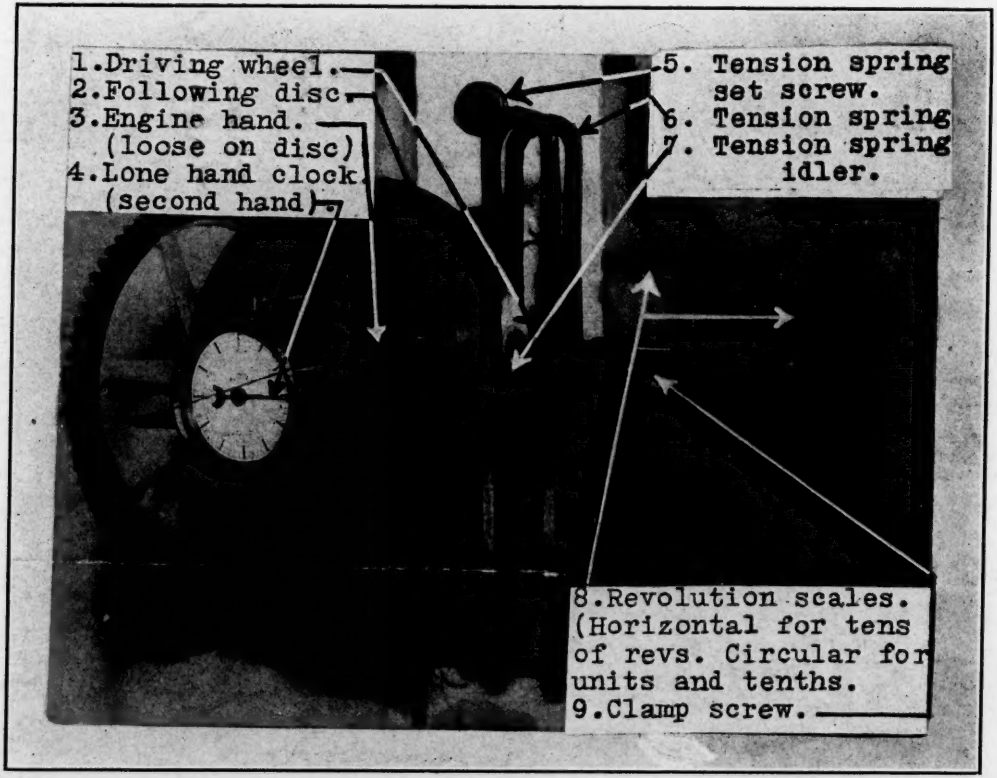
$$S = \frac{10 \times 60 \times 3}{R \times P} = \frac{1{,}800}{R \times P},$$

where

R = Revolutions per minute at the time;
P = Pitch of propeller in feet;
S = Seconds.

Upon consideration, it will be perceived that this procedure (of making the *engine room* gain or lose distance) is a *natural* process in the work of maintaining position. The bridge determines by observation the distance to be gained or lost, and *distance* is an integral function of speed; the *source* of power to make speed or distance resides in the engine room. The bridge, in estimating the number of revolutions to gain or lose distance, deliberately or unconsciously performs a mental process which can be performed to better advantage in the engine room; first, because it can be worked out in tabular form; and, second, for the practical consideration that the engine room operates the motive power which is to produce the desired results. The *bridge* should be divested of all activities which can be performed elsewhere, especially if to better advantage.

The following extract from the report of a Board composed of senior engineer officers on the armored cruiser squadron of the Pacific Fleet gives a clear idea of the construction of the first instrument installed on the *Washington* some years ago. "The device consists essentially of a 12-inch flat disc of steel driven by a 4-inch wheel perpendicular to its


1.Driving wheel.
2.Following disc.
3.Engine hand. (loose on disc)
4.Lone hand clock. (second hand)
5. Tension spring set screw.
6. Tension spring
7. Tension spring idler.
8.Revolution scales. (Horizontal for tens of revs. Circular for units and tenths.
9.Clamp screw.

face and which is driven by a shaft connected by small gear wheels to the starboard revolution indicator. The distance of the driving wheel from the center of the large disc can be varied by a screw of ½-inch pitch. On this screw is placed a scale of revolutions. A clock of 4-inch dial with a lone second hand is placed over the center of the large disc, the latter carrying a U, over the ends of which an elastic band is stretched across the face of the clock. When the revolutions of the main engines agree with the pointer on the scale the disc should make one revolution per minute, the U pointer exactly following the clock hand, and, conversely, by moving the scale indicator until the U pointer exactly follows the clock hand the number of revolutions of the main engine can be read from the scale.

"The device on board the *Washington* has been constructed and installed by the ship's force under the direction of Lieutenant Commander Kaiser, the castings being furnished from navy yards. It has been in use for nearly a year, during which time the ship has cruised 20,000 miles, generally in squadron, both as guide and following ship. The operation of changing the index to agree with deck signals can be made in an instant, and experience shows that even green men can be quickly trained to make the disc hand follow the clock hand, thus insuring regularity of speed. An error in the clock rate would make a small error in the exactness of the total revolutions, but they would still be uniform, which is most to be desired.

"The device is stoutly constructed, with no delicate parts and few wearing parts that would ever require renewal. The large disc is pressed against its driving disc, which is knurled on the circumference by a U spring of variable tension with an idle wheel, so that there appears to be no slip. The large disc would probably become grooved and rough on its driving face, requiring renewal perhaps once a year. (See photograph.)

"The Board is of the opinion that constant and regular revolutions can be maintained by this device and that it is

particularly valuable to the guide ship; that it materially shortens the time of adjusting the revolutions to either large or small changes of speed, thus making it valuable to following ship; its scale is a constant index of the number of revolutions being made; it shows the officer on watch in the engine room at a glance whether the revoultions have been as desired for a past interval as it shows the accumulated error."

An instrument was built from *Washington* blue-prints and installed on the *Connecticut* during the battleship cruise around the world. The fleet engineer reported: "The cruise of the fleet has demonstrated the necessity for some instrument which will show the exact speed of revolutions of the engines at any time. Such an instrument would make it possible for the engines to be run at constant speed, and in changes of speed enable the man at the throttle to take up new speed with practically no loss of time in counting revolutions, as at present.

"The only error noted up to date has been so small as to be practically negligible, due to the inaccuracy of the two-dollar lone hand clock used with the tachometer."

The character of the traction surfaces—the surface of the disc and the rim of the driving wheel—has an important bearing on the correct functioning of this instrument. It has been found by experience that best results are obtained by making the disc of a saw-steel blank and making the *rim* of the driving wheel of small radius and of hardened steel. In pouring a stream of oil over the point of contact and also in coating the face of the disc with lard no *slip* was discovered. If the *disc* is soft and if the *rim* is *broad* the contact *point* becomes a *line* of *contact* in a radial direction on the disc; the outer and inner radial points travel at different *linear* speeds. Setting up on the tension device therefore only aggravates the difficulty. This occurred on some instruments originally supplied. The remedy is simple.

It was apprehended by some that the driving wheel would wear a groove in the disc if the engine were run a long time at one speed. In an instrument constructed on board the

Montgomery, from parts supplied by Cory & Son, the maximum error found when set *one-tenth* of *one turn* on either side of 105 (at which the engine had been run for a large number of revolutions) was one-sixth of 1 per cent.; the average was much smaller. In this instrument the disc had been made half size (as compared to that on the *Washington*), so the error would be doubled. Experience shows that a disc of 6-inch or even 4-inch radius is large enough.

With an A. C. generator driven by the main shaft (or the averaging shaft of the engine revolution counter) and a synchronous motor which will start from rest and keep step, without hunting, a reliable means is supplied by which the speed condition in the engine room can be reproduced on the bridge and in the central station or elsewhere. Such a power-transmission device is understood to have been developed. It is highly desirable that an efficient speed control should be provided, especially in our battleships, by which the "caterpillar" movement down the column may be minimized; it is, moreover, a tactical *necessity* of prime importance, and of growing importance, if high-speed ships of large tonnage are to maneuver at close distance efficiently and with safety.

SAFETY-VALVE SPRINGS.

By Wm. L. DeBaufre, Mechanical Engineer, Associate.

Of each lot of safety valves purchased for the ships of the Navy two are sent for test to the Naval Engineering Experiment Station at Annapolis, Maryland, and are mounted in turn on an oil-burning marine boiler for observation of their general performance. From time to time the discharge capacities of the various makes of valves are also measured while the valve under test is blowing steadily for two hours. After dismounting the valve from the boiler, each spring is removed and compressed in a testing machine to determine the relation between the load and the spring compression. The spring is also compressed solid to show the general character of the spring material by the set resulting therefrom. From the dimensions of the spring are calculated the fiber stress for the load corresponding to the maximum lift of the valve and the shearing modulus of elasticity of the spring material.

In this article will be explained the method used at the above Station of testing safety-valve springs and the application of the proper formulas to the calculation of the fiber stress. Since questions have at times arisen regarding the design of safety valves to meet the Navy requirements, there will also be explained how to predetermine the proper spring dimensions for given safety-valve conditions. Finally, there will be given the derivation of the formulas used.

METHOD OF TESTING.

Before mounting in the testing machine, the dimensions of a safety-valve spring, such as is shown in Fig. 1, are measured with micrometers. The bar which was originally $1\frac{1}{8}$ inches

square in this case, has been forced into the trapezoidal section seen in Fig. 2, by winding it around a $3\frac{9}{16}$-inch diameter mandrel to form the helical spring. The spring has an internal diameter about $\frac{1}{16}$-inch greater than that of the mandrel. To serve in accurately measuring the inside diameter of the coils of helical springs the cast-iron plate in Fig. 1 was turned with concentric grooves to center the spring with respect to a rod

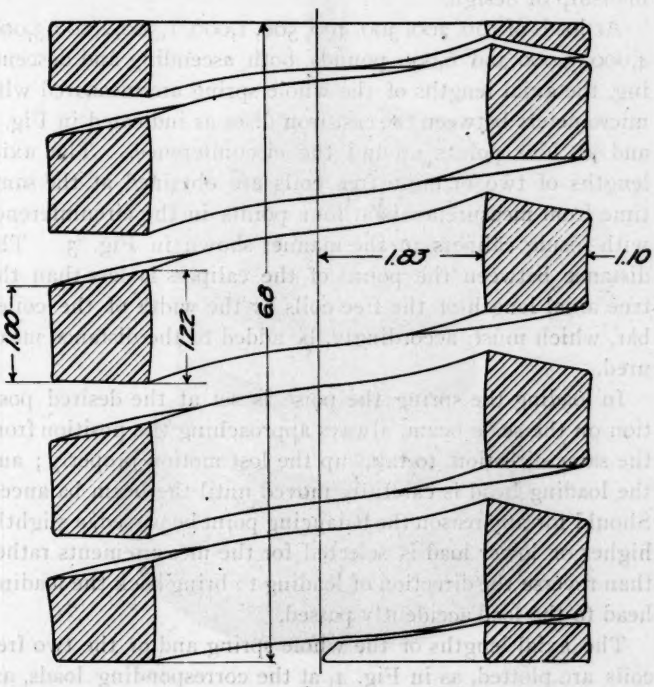


FIG. 2.

screwed into the plate perpendicular to its face. Four measurements for each coil are made along diameters at right angles and from the middle points of the inside face of the coil to the surface of the rod. The inside radius of the spring, as marked in Fig. 2, is equal to the average of these measurements plus the radius of the rod.

In the testing machine the spring is mounted between the two cast-iron discs seen in Fig. 8. These discs are made with spherical end surfaces to allow the spring to take its natural position under the various loads. In the majority of safety valves used in the naval service the spring supports are made to allow free movement of the spring. In some instances, however, the free movement may be prevented by poor workmanship or design.

At loads of 100, 200, 300, 400, 500, 1,000, 1,500, 2,000, 3,000, 4,000, 5,000 and 6,000 pounds, both ascending and descending, the axial lengths of the whole spring are measured with micrometers between the cast-iron discs as indicated in Fig. 3 and at four points around the circumference. The axial lengths of two or more free coils are obtained at the same time from measurements at four points in the circumference with inside calipers in the manner shown in Fig. 3. The distance between the points of the calipers is less than the true axial length of the free coils by the width of the coiled bar, which must, accordingly, be added to the distance measured.

In loading the spring the poise is set at the desired position on the scale beam, always approaching the position from the same direction, to take up the lost motion properly; and the loading head is carefully moved until the beam balances. Should for any reason the balancing point be passed, a slightly higher or lower load is selected for the measurements rather than reverse the direction of loading to bring back the loading head to the load accidently passed.

The axial lengths of the whole spring and of the two free coils are plotted, as in Fig. 4, at the corresponding loads, ascending points being marked with circles and descending points with crosses. Straight lines are drawn through the points plotted. The departure from the straight line of the points plotted for the length of the whole spring at loads less than 500 pounds is due to the formation of the ends of the spring to secure a solid bearing. The similar departure from the straight line of the points plotted for the two free coils

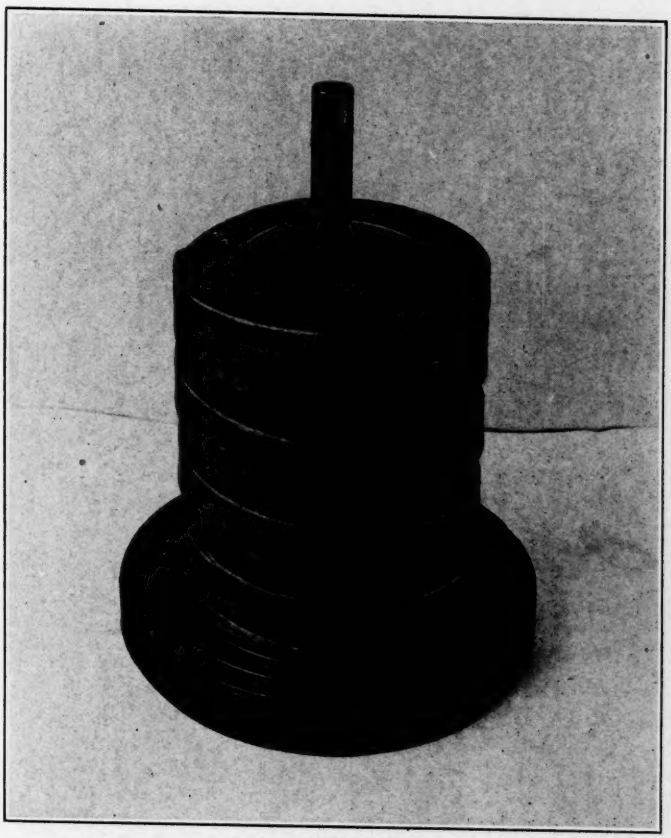


FIG. 1.

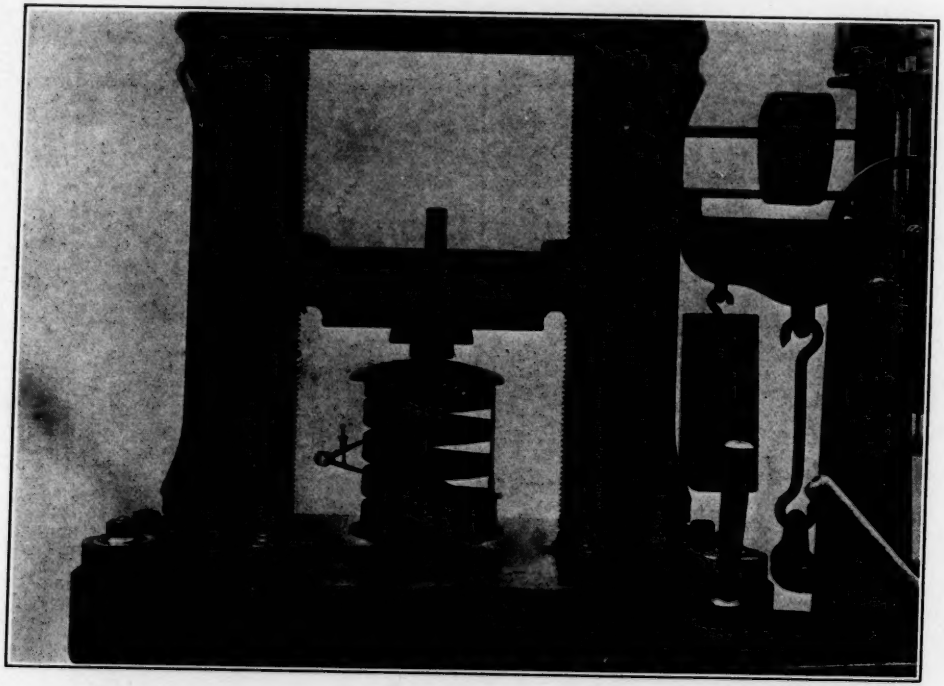


FIG. 3.

shows that in this case the coils selected for measurement were too near the ends of the spring, and that the measurements are accordingly slightly in error.

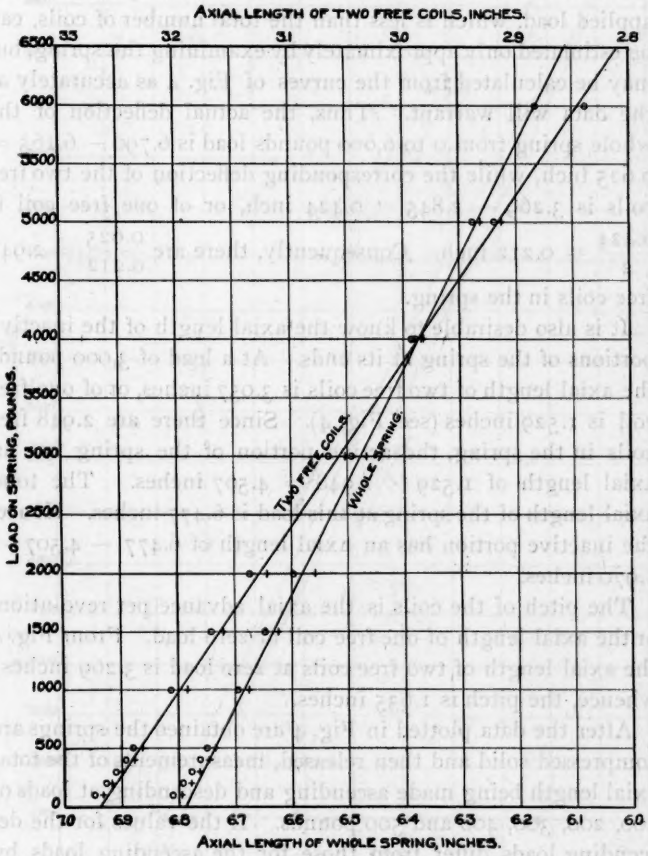


FIG. 4.

Above 500 pounds, however, the axial deflection of the spring is proportional to the load, and the stiffness of the spring may be measured by the increase in load per inch of axial deflection, called the modulus of stiffness. For the spring

tested, the modulus of stiffness is $\frac{6{,}000 - 0}{6.790 - 6.165} = 9{,}600$ pounds per inch.

The number of free coils available for actively resisting the applied load, which is less than the total number of coils, can be estimated only approximately by examining the spring, but may be calculated from the curves of Fig. 4 as accurately as the data will warrant. Thus, the actual deflection of the whole spring from 0 to 6,000 pounds load is $6.790 - 6.165 = 0.625$ inch, while the corresponding deflection of the two free coils is $3.269 - 2.845 = 0.424$ inch, or of one free coil is $\frac{0.424}{2} = 0.212$ inch. Consequently, there are $\frac{0.625}{0.212} = 2.948$ free coils in the spring.

It is also desirable to know the axial length of the inactive portions of the spring at its ends. At a load of 3,000 pounds the axial length of two free coils is 3.057 inches, or of one free coil is 1.529 inches (see Fig. 4). Since there are 2.948 free coils in the spring, the active portion of the spring has an axial length of $1.529 \times 2.948 = 4.507$ inches. The total axial length of the spring at this load is 6.477 inches. Hence, the inactive portion has an axial length of $6.477 - 4.507 = 1.970$ inches.

The pitch of the coils is the axial advance per revolution, or the axial length of one free coil at zero load. From Fig. 4 the axial length of two free coils at zero load is 3.269 inches; whence, the pitch is 1.635 inches.

After the data plotted in Fig. 4 are obtained the springs are compressed solid and then released, measurements of the total axial length being made ascending and descending at loads of 100, 200, 300, 400 and 500 pounds. If the values for the descending loads differ from those for the ascending loads by five-hundredths (0.05) inch, the spring is of poor material and is rejected. The curve in Fig. 5 has been plotted simply to show the performance of a spring which is compressed solid and then released. The fiber stress in the material when compressed solid amounted in this case to 87,000 pounds per

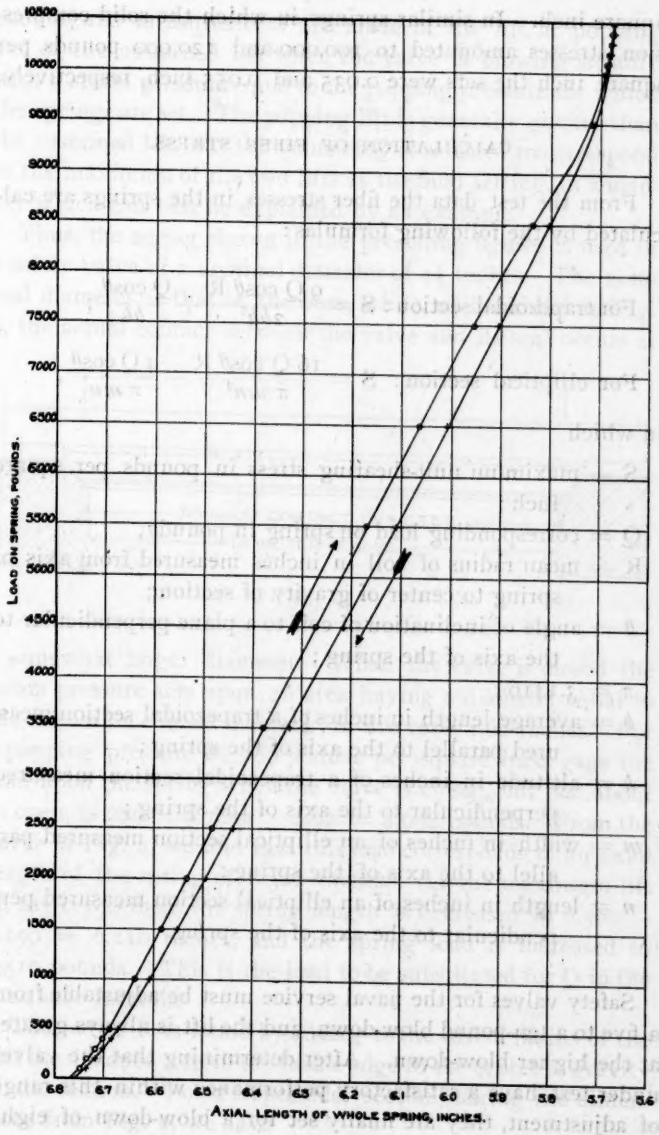

LOAD ON SPRING, POUNDS.
10500
10000
9500
9000
8500
8000
7500
7000
6500
6000
5500
5000
4500
4000
3500
3000
2500
2000
1500
1000
500
0
6.8
6.7
6.6
6.5
6.4
6.3
6.2
6.1
6.0
5.9
5.8
5.7
5.6
AXIAL LENGTH OF WHOLE SPRING, INCHES.

FIG. 5.

square inch. In similar springs in which the solid compression stresses amounted to 100,000 and 120,000 pounds per square inch the sets were 0.035 and 0.055 inch, respectively.

CALCULATION OF FIBER STRESS.

From the test data the fiber stresses in the springs are calculated by the following formulas:

$$\text{For trapezoidal section: } S = \frac{9\,Q\cos\theta\,R}{2bh^2} + \frac{Q\cos\theta}{bh};$$

$$\text{For elliptical section: } S = \frac{16\,Q\cos\theta\,R}{\pi\,mn^2} + \frac{4\,Q\cos\theta}{\pi\,mn},$$

in which

S = maximum unit-shearing stress in pounds per square inch;
Q = corresponding load on spring in pounds;
R = mean radius of coil in inches measured from axis of spring to center of gravity of section;
θ = angle of inclination of coil to a plane perpendicular to the axis of the spring;
π = 3.1416;
b = average length in inches of a trapezoidal section measured parallel to the axis of the spring;
h = altitude in inches of a trapezoidal section measured perpendicular to the axis of the spring;
m = width in inches of an elliptical section measured parallel to the axis of the spring;
n = length in inches of an elliptical section measured perpendicular to the axis of the spring.

Safety valves for the naval service must be adjustable from a five to a ten-pound blow-down, and the lift is always greater at the higher blow-down. After determining that the valves under test have a satisfactory performance within this range of adjustment, they are finally set for a blow-down of eight

pounds, and measurements are made of the lift at popping and of the sustained lift when the valve is blowing steadily with a steam pressure equal to the popping pressure for which the springs are set. The popping lift is generally greater than the sustained lift, but the value of Q is selected to correspond to the maximum of the two lifts at the final setting, in a manner which can best be explained by an example.

Thus, the spring shown in the preceding figures is used in a safety valve of a nominal diameter of $4\frac{1}{2}$ inches. The nominal diameter is that of the bore, while, as indicated in Fig. 6, the actual contact between the valve and its seat occurs at

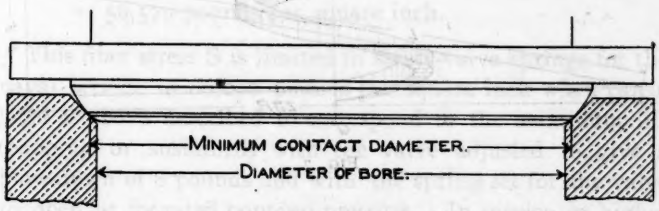


FIG. 6.

a somewhat larger diameter. When the valve is closed the steam pressure acts upon an area having a diameter equal to the minimum contact diameter, in this case, 4.62 inches. For a popping pressure of 280 pounds per square inch gage the load upon the spring when the valve is closed, but just about to open, is $280 \times 0.7854 \times 4.62^2 = 4{,}694$ pounds. From the curve of Fig. 4, we find that this load corresponds to an axial length of the spring of 6.300 inches. At the maximum lift of, say 0.190 inch, the spring length is decreased to $6.300 - 0.190 = 6.110$ inches, and the spring load is increased to 6,510 pounds. This is the load to be substituted for Q in the the above formula.

The radius R is found by adding to the inside radius of the coil the distance x from the inside edge to the center of gravity of the section. For a trapezoidal section, taking the dimensions from Fig. 2, we have

$$x = \frac{h}{3}\frac{2b_1 + b_2}{b_1 + b_2},$$

$$= \frac{1.10}{3}\frac{2 \times 1.00 + 1.22}{1.00 + 1.22},$$

$$= 0.532 \text{ inch.}$$

Hence,

$$R = 1.83 + 0.53 = 2.36 \text{ inches.}$$

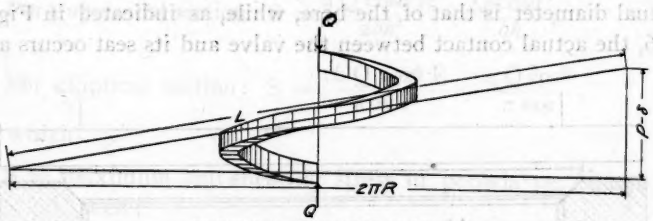


FIG. 7.

By referring to Fig. 7, the basis of determining the cos θ may be made clear. If we imagine one of the coils unwrapped while under the load Q, it will form the hypothenuse L of a right triangle of which the base is the circumference of a cylinder of radius R, and the altitude of the triangle is the pitch p minus the deflection δ of one coil under the load Q. Thus

$$L = \sqrt{(p - \delta)^2 + (2\pi R)^2} \text{ and } \cos\theta = \frac{2\pi R}{L}.$$

The pitch p minus the deflection δ is the axial length of a free coil under the load Q; and from Fig. 4 we find this axial length under a load of 6,510 pounds to be $(p - \delta) = \frac{2.808}{2}$ = 1.404 inches. Since R = 2.36 inches, we obtain

$$L = \sqrt{(1.404)^2 + (2 \times 3.1416 \times 2.36)^2},$$

$$= 14.89 \text{ inches,}$$

and,

$$\cos\theta = \frac{2 \times 3.1416 \times 2.36}{14.89},$$
$$= 0.9959.$$

Substituting $b = \frac{b_1 + b_2}{2} = \frac{1.00 + 1.22}{2} = 1.11$ inches, $h = 1.10$ inches, and the above values in the formula for the trapezoidal section, we obtain

$$S = \frac{9 \times 6{,}510 \times 0.9959 \times 2.36}{2 \times 1.11 \times 1.10^2} + \frac{6{,}510 \times 0.9959}{1.221},$$
$$= 51{,}260 + 5{,}310,$$
$$= 56{,}570 \text{ pounds per square inch.}$$

This fiber stress S is limited in safety-valve springs for the naval service to 60,000 pounds per square inch when calculated as above described to correspond to the maximum lift (popping or sustained) with the valve adjusted to give a blow-down of 8 pounds and with the spring set for the valve to open at its rated popping pressure. In service, at higher blow-downs or higher popping pressures, the stress calculated as above will be exceeded. For this reason and because it is considered wise to be amply safe in a device upon which the safety of the crew and ship depends, all springs giving a stress higher than 60,000 pounds per square inch under the above conditions, are rejected.

CALCULATION OF SHEARING MODULUS OF ELASTICITY.

There are also calculated for the springs tested, the values of the shearing modulus of elasticity of the material. The following formulas are used:

For trapezoidal section:

$$\delta = \frac{3(b^2 + h^2)}{b^3h^3} \frac{Q \cos^2\theta \, R^2 L}{0.843 G} + \frac{1}{bh} \frac{Q \cos^2\theta \, L}{G},$$

For elliptical section:

$$\delta = \frac{16(m^2 + n^2)}{\pi m^3 n^3} \frac{Q \cos^2\theta \, R^2 L}{G} + \frac{4}{\pi mn} \frac{Q \cos^2\theta \, L}{G}.$$

in which

δ = axial deflection in inches of one free coil;
Q = corresponding load on spring in pounds;
R = means radius of coil in inches measured from axis of spring to center of gravity of section;
L = true length in inches of one free coil;
θ = angle of inclination of coil to a plane perpendicular to the axis of the spring;
π = 3.1416;
b = average length in inches of a trapezoidal section measured parallel to the axis of the spring;
h = altitude in inches of a trapezoidal section measured perpendicular to the axis of the spring;
m = width in inches of an elliptical section measured parallel to the axis of the spring;
n = length in inches of an elliptical section measured perpendicular to the axis of the spring;
G = shearing modulus of elasticity of the material in pounds per square inch.
0.843 = constant for rectangular section (see page 290).

From Fig. 4 we find the deflection of two free coils under a load of 6,000 pounds to be 3.269 − 2.845 = 0.424 inch, whence δ = 0.212 inch for Q = 6,000 pounds. From the calculations on page 276 we have R = 2.36 inches and R^2 = 5.570. Referring to Fig. 7 we have

$$L = \sqrt{(p - \delta)^2 + (2\pi R)^2}.$$

At 6,000 pounds the axial length of two free coils is 2.845 inches, whence $(p - \delta)$ = 1.423 inches. Substituting in the above expression,

$$L = \sqrt{(1.423)^2 + (2 \times 3.1416 \times 2.36)^2},$$
$$= 14.90 \text{ inches.}$$

$$\text{Now, } \cos^2\theta = \left[\frac{2\pi R}{L}\right]^2,$$
$$= \left[\frac{2 \times 3.1416 \times 2.36}{14.90}\right]^2,$$
$$= 0.9904.$$

Substituting $b = \frac{1.00 + 1.22}{2} = 1.11$ inches, $h = 1.10$ inches, and the above values in the formula for a trapezoidal section, we obtain

$$0.212 = \frac{3(1.11^2 + 1.10^2)}{1.11^3 \times 1.10^3} \frac{6{,}000 \times 0.9904 \times 5.570 \times 14.90}{0.843 \times G} + \frac{1}{1.11 \times 1.10} \frac{6{,}000 \times 0.9904 \times 14.90}{G},$$

$$= \frac{2{,}354{,}600}{G} + \frac{72{,}500}{G};$$

$$G = 11{,}107{,}000 + 342{,}000,$$

$= 11{,}449{,}000$ pounds per square inch, shearing modulus of elasticity of the material.

While the values found for the shearing modulus of elasticity of a number of safety-valve springs has varied considerably, a fair average valve for design purposes is apparently 11,750,000 pounds per square inch.

SAFETY-VALVE SPRING DESIGN.

The formulas used above for determining the working stress and the modulus of elasticity of safety-valve springs were derived particularly to calculate these values for a spring already manufactured and under test. For design purposes the calculations may be somewhat simplified by neglecting the effect of the inclination of the coils and by substituting in the formulas the dimensions of the original bar in place of those of the deformed section. The error in so doing may be shown by the following calculations.

The spring in Fig. 2 was made of a bar originally $1\frac{1}{8}$-inch square by winding it on a mandrel $3\frac{9}{16}$ inches diameter. We will assume the spring to have an inside diameter $\frac{1}{16}$-inch larger than that of the mandrel.

For a square section, eliminating $\cos\theta$, we have

$$S = \frac{9\,Q\,R}{2b^3} + \frac{Q}{b^2}.$$

Substituting $Q = 6{,}510$ pounds,
$b = 1.125$ inches,

$$R = \frac{3.625 + 1.125}{2} = 2.375 \text{ inches};$$

we obtain $$S = \frac{9 \times 6{,}510 \times 2.375}{2 \times 1.125^3} + \frac{6{,}510}{1.125^2}$$
$$= 48{,}865 + 5{,}144 = 54{,}009 \text{ pounds per square inch.}$$

This stress is about 2.5 per cent. less than that obtained on page 277 for the actual spring, thus indicating that an allowance of at least three per cent. should be made when the calculations are based upon the dimensions of the original bar. We also have

$$\delta = \frac{6\,Q\,R^2\,L}{b^4\,0.843\,G} + \frac{1}{b^2}\frac{Q\,L}{G}.$$

Substituting $Q = 6{,}000$ pounds,
$b = 1.125$ inches,

$$R = \frac{3.625 + 1.125}{2} = 2.375 \text{ inches},$$

$L = 2\pi R = 14.92$ inches,
$G = 11{,}750{,}000$ pounds per square inch;

we obtain

$$\delta = \frac{6}{1.125^4}\frac{6{,}000 \times 2.375^2 \times 14.92}{0.843 \times 11{,}750{,}000} + \frac{1}{1.125^2}\frac{6{,}000 \times 14.92}{11{,}750{,}000}$$
$$= 0.1910 + 0.0060$$
$$= 0.1970 \text{ inch, about 7 per cent. less than the value 0.212 inch given on page 278 for the actual spring.}$$

To show the application of these formulas to design purposes let the problem be given to design a safety-valve spring

to meet the Navy requirements for a four-inch safety valve which is to have a popping pressure of 265 pounds gage. To solve this problem it is necessary to obtain certain empirical data regarding the type of valve for which the spring is to be designed.

When a safety valve is closed, but just about to open, the load on the spring is equal to the product of the steam pressure and the area within the diameter of contact between the valve disc and its seat. When the valve opens an additional pressure is exerted by the steam upon the disc, due either to a change in the direction of its flow caused by a curved lip on the valve disc, as in Fig. 8, or to a restriction in the flow by a huddling ring, as in Fig. 9. It may be mentioned that the

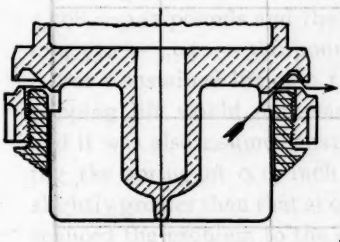


FIG. 8.

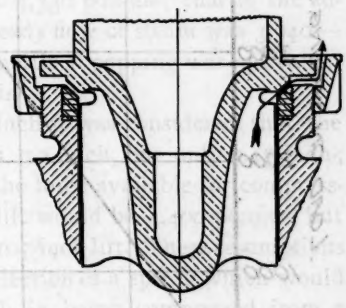


FIG. 9.

flow of steam through the throat formed between the disc and seat will be unaffected by any further restrictions provided the pressure in the huddling chamber does not exceed about one-half the initial pressure. The additional pressure exerted by the flowing steam holds the valve open at a certain "sustained lift" while it is steadily blowing. At popping, however, a greater lift may be momentarily attained, due to inertia of the moving parts which carries them beyond the point of equilibrium. With the type of huddling device shown in Fig.

8 the popping lift may exceed the sustained lift by one hundred per cent., while, with the device shown in Fig. 9, the sustained lift may even slightly exceed the popping lift. This relation must be known for the type of valve under considera-

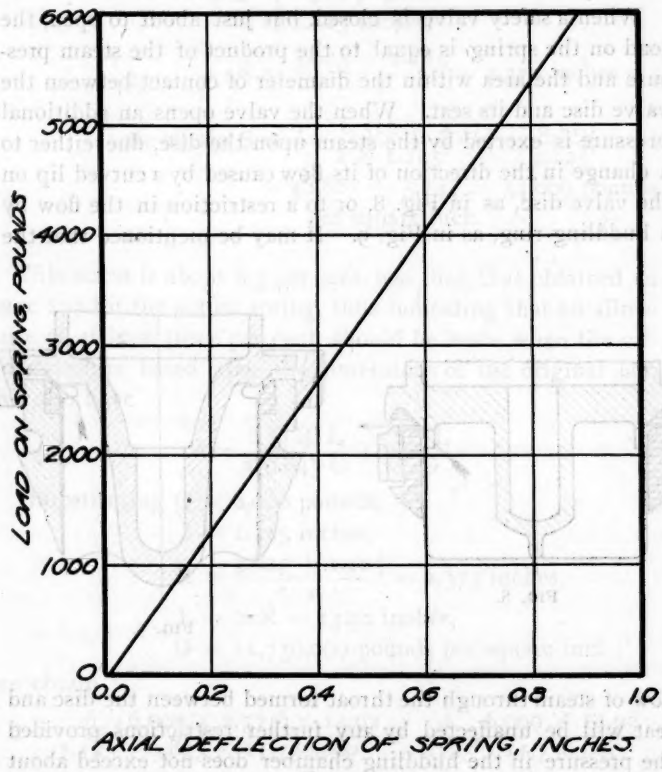


FIG. 10.

tion before attempting to design a spring to give a sustained lift of one-tenth inch, as has been required by the Navy specifications, and yet not exceed the fiber stress specified. It is also necessary to know how much pressure additional to that exerted by the steam upon the closed valve is available to

compress the spring when the valve lifts at popping. This can only be determined by experiments such as are made at the Naval Engineering Experiment Station during acceptance tests of valves for the naval service.

The test of a particular 4-inch valve of the type shown in Fig. 8 gave a popping lift of 0.107 inch and a sustained lift of 0.054 inch, and a test of the spring later gave the curve of Fig. 10. With the valve closed, the load upon the spring would be $265 \times 0.7854 \times 4.165^2 = 3{,}598$ pounds, the contact diameter of 4.165 inches being found by measurement on this 4-inch valve. The spring deflection for Fig. 10, corresponding to the load of 3,598 pounds, is 0.531 inch. Adding to this deflection the further deflections of 0.054 inch and 0.107 inch, respectively, we may obtain from Fig. 10 the corresponding loads of 3,940 pounds and 4,330 pounds; that is, the additional pressure due to the steady flow of steam was $3{,}940 - 3{,}598 = 342$ pounds and the inertia at popping was equivalent to $4{,}330 - 3{,}940 = 390$ pounds.

For a sustained lift of 0.1 inch it was considered that the popping lift would not reach 0.2 inch, but only 0.17 inch; and it was also assumed that the force available for compressing the spring at 0.17-inch lift would be 4,400 pounds, but slightly greater than that at 0.107-inch lift. These assumptions reduced the problem to the selection of a spring which would give a deflection of 0.17 inch in being compressed from a load of 3,600 pounds to a load of 4,400 pounds, the fiber stress at the higher load not to exceed 60,000 pounds per square inch.

In the formula

$$S = \frac{9\,Q\,R}{2\,b^3} + \frac{Q}{b^2}$$

we accordingly have $Q = 4{,}400$ pounds and $S = 57{,}000$ pounds per square inch (less than the 60,000 pounds per square inch specified to allow for the deformation of the spring section in winding). Substituting several values for the size b of the bar, we obtain the corresponding inside diameter of the spring from $(2\,R - b)$. These valves are tabulated below:

Size of bar, inches.	Inside diameter of spring, inches.
$\frac{5}{8}$	0.503, say $\frac{1}{2}$
$\frac{11}{16}$	0.878, say $\frac{7}{8}$
$\frac{3}{4}$	1.346, say $1\frac{5}{16}$
$\frac{13}{16}$	1.915, say $1\frac{7}{8}$
$\frac{7}{8}$	2.593, say $2\frac{1}{2}$
$\frac{15}{16}$	3.390, say $3\frac{1}{4}$
1	4.313, say $4\frac{1}{4}$

A smaller inside diameter may be used for a given size of bar, but the design will not be as economical.

Let δ_1 be the deflection per coil of the spring when the valve is closed and δ_2 the deflection per coil with the valve opened at its maximum lift. Let Q_1 and Q_2 be the corresponding spring loads. Then

$$\delta_1 = \frac{6}{b^4} \frac{Q_1 R^2 L}{0.843 G} + \frac{1}{b^2} \frac{Q_1 L}{G},$$

and

$$\delta_2 = \frac{6}{b^4} \frac{Q_2 R^2 L}{0.843 G} + \frac{1}{b^2} \frac{Q_2 L}{G},$$

whence,

$$\delta_2 - \delta_1 = \left[\frac{6}{b^4} \frac{R^2 L}{0.843 G} + \frac{1}{b^2} \frac{L}{G} \right] (Q_2 - Q_1).$$

The number of free coils is obtained by dividing the deflection desired by the numerical value of $(\delta_2 - \delta_1)$.

For the sizes of bar and inside diameters tabulated above we obtain for $(\delta_2 - \delta_1) = 0.17$ inch and $(Q_2 - Q_1) = 800$ pounds, the deflection per free coil and the number of free coils as below:

Size of bar, inches.	Inside diam. of spring, inches.	Deflection per coil, inch.	Number of free coils.
$\frac{5}{8}$	$\frac{1}{2}$	0.004167	40.79
$\frac{11}{16}$	$\frac{7}{8}$	0.007206	23.59
$\frac{3}{4}$	$1\frac{5}{16}$	0.011338	14.99
$\frac{13}{16}$	$1\frac{7}{8}$	0.017823	9.54
$\frac{7}{8}$	$2\frac{1}{2}$	0.025904	6.56
$\frac{15}{16}$	$3\frac{1}{4}$	0.037197	4.57
1	$4\frac{1}{4}$	0.056197	3.03

The first three springs may be eliminated by reason of their great length, due to the number of free coils required. The last two springs have outside diameters of $5\frac{1}{8}$ and $6\frac{1}{4}$ inches, respectively, and if these diameters are too large to be accommodated in the valve head, the choice is further restricted to the $\frac{13}{16}$ and $\frac{7}{8}$-inch sizes. The volumes of these two springs are $3.1416 \times (1\frac{7}{8} + \frac{13}{16}) \times 9.54 \times (\frac{13}{16})^2 = 53.17$ cubic inches and $3.1416 \times (2\frac{1}{2} + \frac{7}{8}) \times 6.56 \times (\frac{7}{8})^2 = 53.25$ cubic inches, respectively, thus showing no advantage in weight of one spring over the other. The first spring may be slightly more costly than the second by reason of the greater number of coils required. We will select the second spring because it will result in the valve requiring less head room than if the first spring were selected.

The deflection per coil of this spring ($b = \frac{7}{8}$-inch) under the load of 4,400 pounds will be found by substituting in

$$\delta_2 = \frac{6\,Q_2 R^2 L}{b^4\, 0.843\, G} + \frac{1\,Q_2 L}{b^2 G}$$

to be $\delta_2 = 0.1892$ inch. Consequently, the pitch must be at least $\frac{7}{8} + 0.1892 = 1.0642$ inch, plus an allowance for the deformation of the spring section. The pitch should not be so great that in compressing the spring solid the elastic limit of the material will be exceeded and the spring receive a permanent set. For good spring material properly tempered a fiber stress of about 90,000 pounds per square inch when compressed solid is considered permissible. Since both the deflection and the maximum fiber stress in a safety-valve spring are directly proportional to the load, we may obtain the deflection at say 85,000 pounds per square inch fiber stress by the proportion

$$\frac{\delta}{0.1892} = \frac{85{,}000}{57{,}000}$$

whence $\delta = 0.282$ inch. This corresponds to a pitch of $0.282 + \frac{7}{8} + (\text{say})\ 0.093 = 1.25$ or $1\frac{1}{4}$ inch, allowing 0.093 inch for the deformation of the section of the bar.

The axial length of the active portion of the spring will be $1.25 \times 6.56 = 8.20$ inches. Assuming an inactive length of twice the bar size, we find the total axial length of the spring will be $8.20 + 2 \times \frac{1}{8} = 9.95$, say 10 inches. The substitution of this spring in the valve giving the test data on page 283 will result in the attainment of the desired lift with a fiber stress within the value specified, provided the huddling device is adapted to the higher lifts. It may be found that the valve action at the higher lifts will be unsatisfactory due to chattering, and the huddling device must then be modified. In the particular case for which the test data on page 283 were obtained, chattering did occur when a more flexible spring was installed, but satisfactory valve action was finally obtained with increased lift by modifying the huddling device from the form in Fig. 8 to a form similar to that in Fig. 9.

To aid in the selection of safety-valve springs to meet the naval requirements Table I has been prepared, for square sections. The values of the load of Q pounds and the deflection per coil of δ inches correspond to a maximum fiber stress just within 60,000 pounds per square inch for the actual spring. The pitch of p inches is selected to give a maximum fiber stress with the spring compressed solid of about 90,000 pounds per square inch. The values are tabulated to correspond to various inside diameters of D inches and various breadths b inches of square bars before coiling. The mandrel upon which the spring is wound should be slightly less in diameter than the tabulated value of D. The method of using this table is made evident by the preceding calculations.

DERIVATION OF FORMULAS.

With the ends of a safety-valve spring formed as in Fig. 1 and supported as usual in marine safety valves the load on the spring is equivalent to a single force acting along the axis, and the formulas for the fiber stress and for the spring deflection may be derived upon this basis. In Fig. 11 the ends of

		1	1-1/4	1-1/2	1-3/4	2	
b	1/8	41.91 0.1171 0.32	34.59 0.1760 0.40	29.44 0.2470 0.51	25.63 0.3301 0.64	22.69 0.4252 0.78	
	3/16	131.4 0.0859 0.34	109.8 0.1267 0.40	94.30 0.1755 0.47	82.63 0.2326 0.56	73.54 0.2975 0.66	
	1/4	290.8 0.0707 0.39	245.7 0.1026 0.44	212.7 0.1404 0.49	187.5 0.1843 0.56	167.6 0.2342 0.63	
	5/16	532.7 0.0620 0.44	454.4 0.0884 0.48	396.2 0.1192 0.53	351.2 0.1557 0.59	315.4 0.1966 0.65	
	3/8	866.5 0.0564 0.51	745.6 0.0793 0.54	654.3 0.1061 0.58	582.9 0.1369 0.63	525.6 0.1718 0.68	
	7/16	1300 0.0528 0.57	1127 0.0730 0.60	995.1 0.0967 0.64	890.6 0.1238 0.68	808.0 0.1543 0.72	
	1/2	1839 0.0502 0.64	1606 0.0685 0.67	1425 0.0898 0.70	1281 0.1141 0.73	1163 0.1415 0.77	
	9/16	2488 0.0483 0.71	2186 0.0652 0.73	1950 0.0847 0.76	1760 0.1068 0.79	1603 0.1316 0.83	
	5/8	3250 0.0471 0.77	2873 0.0627 0.80	2574 0.0808 0.82	2331 0.1012 0.85	2131 0.1240 0.89	
	11/16	4130 0.0463 0.84	3670 0.0608 0.86	3302 0.0777 0.89	3001 0.0967 0.92	2750 0.1179 0.95	
	3/4	5130 0.0456 0.91	4580 0.0595 0.93	4137 0.0753 0.96	3772 0.0931 0.98	3466 0.1129 1.01	
	13/16	6251 0.0453 0.98	5607 0.0584 1.00	5082 0.0734 1.02	4648 0.0902 1.05	4282 0.1089 1.08	
	7/8	7496 0.0451 1.05	6751 0.0576 1.07	6140 0.0719 1.09	5631 0.0878 1.12	5200 0.1055 1.14	
	15/16	8867 0.0450 1.12	8016 0.0570 1.14	7313 0.0707 1.16	6724 0.0859 1.18	6223 0.1027 1.21	
	1	10364 0.0450 1.19	9402 0.0566 1.21	8604 0.0697 1.23	7930 0.0843 1.25	7355 0.1004 1.28	
	1-1/8	13740 0.0453 1.33	12550 0.0562 1.35	11540 0.0684 1.37	10690 0.0819 1.39	9950 0.0968 1.41	
	1-1/4	17640 0.0459 1.48	16190 0.0562 1.49	14970 0.0677 1.51	13920 0.0804 1.53	13000 0.0942 1.55	
	1-1/2	27000 0.0475 1.76	25020 0.0570 1.77	23320 0.0675 1.79	21830 0.0789 1.81	20520 0.0912 1.82	

D = internal diameter of spring in in
Q = load on spring in pounds for fibe
δ = deflection per free coil in inche
p = pitch of coils in inches at no lo

Formulas used: $Q = 57000 \frac{b^3}{4.5\ R + b}$;

TABLE I.

	D											
4	2	2-1/4	2-1/2	2-3/4	3	3-1/4	3-1/2	3-3/4	4	4-1/2	5	
3	22.69	20.36	18.46	16.88	15.56	14.42	13.44	12.59	11.84	10.67	9.551	Q
1	0.4252	0.5323	0.6515	0.7828	0.9261	1.0815	1.2488	1.4283	1.6199	2.0390	2.5065	δ
4	0.78	0.94	1.12	1.31	1.53	1.76	2.01	2.28	2.57	3.20	3.90	p
3	73.54	66.24	60.27	55.28	51.06	47.43	44.29	41.53	39.10	35.00	31.68	Q
5	0.2975	0.3705	0.4516	0.5486	0.6377	0.7428	0.8560	0.9772	1.1064	1.3890	1.7037	δ
5	0.66	0.77	0.89	1.02	1.17	1.33	1.49	1.68	1.87	2.29	2.77	p
5	167.6	151.6	138.4	127.2	117.8	109.6	102.5	96.28	90.76	81.43	73.83	Q
3	0.2342	0.2901	0.3520	0.4200	0.4940	0.5740	0.6601	0.7522	0.8502	1.0645	1.3028	δ
6	0.63	0.72	0.81	0.91	1.02	1.14	1.27	1.41	1.56	1.88	2.24	p
2	315.4	286.2	262.0	241.5	224.0	208.9	195.7	184.0	173.7	156.1	141.8	Q
7	0.1966	0.2422	0.2927	0.3480	0.4082	0.4732	0.5429	0.6175	0.6970	0.8702	1.0628	δ
9	0.65	0.72	0.79	0.87	0.96	1.06	1.17	1.28	1.40	1.66	1.95	p
9	525.6	478.5	439.2	405.9	377.2	352.3	330.5	311.3	294.1	265.0	241.1	Q
9	0.1719	0.2106	0.2535	0.3003	0.3512	0.4061	0.4651	0.5280	0.5950	0.7410	0.9030	δ
3	0.68	0.74	0.80	0.87	0.95	1.03	1.12	1.21	1.31	1.53	1.78	p
6	808.0	736.1	677.3	627.3	584.1	546.5	513.4	484.1	458.0	413.4	376.7	Q
8	0.1543	0.1883	0.2257	0.2666	0.3109	0.3586	0.4098	0.4644	0.5225	0.6489	0.7892	δ
8	0.72	0.77	0.83	0.89	0.96	1.03	1.11	1.19	1.28	1.47	1.68	p
1	1163	1065	982.8	912.0	850.7	797.2	750.0	708.1	670.6	606.4	553.4	Q
1	0.1415	0.1718	0.2061	0.2445	0.2868	0.3232	0.3686	0.4169	0.4684	0.5802	0.7041	δ
3	0.77	0.82	0.87	0.92	0.98	1.05	1.12	1.19	1.27	1.43	1.62	p
0	1603	1472	1361	1266	1183	1110	1046	988.2	936.9	848.7	775.7	Q
8	0.1316	0.1591	0.1893	0.2221	0.2576	0.2958	0.3367	0.3802	0.4264	0.5269	0.6381	δ
9	0.83	0.87	0.92	0.97	1.02	1.08	1.14	1.20	1.27	1.42	1.59	p
1	2131	1962	1818	1693	1585	1489	1405	1329	1262	1145	1048	Q
2	0.1240	0.1492	0.1768	0.2069	0.2393	0.2741	0.3114	0.3510	0.3931	0.4845	0.5884	δ
5	0.89	0.93	0.97	1.01	1.06	1.11	1.17	1.23	1.29	1.43	1.58	p
1	2750	2538	2357	2199	2062	1940	1832	1736	1649	1499	1374	Q
7	0.1179	0.1412	0.1668	0.1945	0.2244	0.2565	0.2908	0.3273	0.3660	0.4499	0.5425	δ
2	0.95	0.99	1.02	1.07	1.11	1.16	1.21	1.26	1.32	1.45	1.59	p
2	3466	3206	2983	2788	2617	2466	2332	2211	2102	1914	1757	Q
1	0.1129	0.1347	0.1586	0.1844	0.2122	0.2420	0.2738	0.3077	0.3435	0.4213	0.5070	δ
8	1.01	1.05	1.08	1.12	1.16	1.21	1.25	1.31	1.36	1.48	1.60	p
8	4282	3969	3699	3463	3256	3072	2907	2760	2626	2395	2201	Q
2	0.1089	0.1294	0.1517	0.1759	0.2020	0.2299	0.2596	0.2912	0.3243	0.3971	0.4770	δ
5	1.08	1.11	1.14	1.18	1.22	1.26	1.30	1.35	1.40	1.51	1.63	p
1	5200	4830	4509	4228	3980	3760	3562	3385	3224	2944	2709	Q
8	0.1055	0.1249	0.1460	0.1688	0.1933	0.2196	0.2476	0.2773	0.3086	0.3786	0.4514	δ
2	1.14	1.17	1.20	1.24	1.27	1.31	1.36	1.40	1.45	1.55	1.66	p
4	6223	5792	5416	5086	4794	4534	4300	4090	3899	3566	3285	Q
9	0.1027	0.1211	0.1412	0.1628	0.1860	0.2108	0.2372	0.2652	0.2949	0.3589	0.4294	δ
8	1.21	1.24	1.27	1.30	1.33	1.37	1.41	1.45	1.50	1.59	1.70	p
0	7355	6857	6423	6040	5700	5396	5124	4877	4653	4262	3931	Q
3	0.1004	0.1180	0.1370	0.1576	0.1796	0.2032	0.2283	0.2549	0.2830	0.3436	0.4039	δ
5	1.28	1.30	1.33	1.36	1.39	1.43	1.47	1.51	1.55	1.64	1.73	p
0	9950	9308	8744	8245	7799	7399	7038	6711	6412	5889	5445	Q
9	0.0968	0.1129	0.1304	0.1492	0.1694	0.1909	0.2137	0.2378	0.2633	0.3183	0.3787	δ
9	1.41	1.44	1.46	1.49	1.52	1.55	1.59	1.62	1.66	1.74	1.83	p
0	13000	12200	11490	10860	10300	9987	9326	8906	8523	7847	7270	Q
4	0.0942	0.1093	0.1255	0.1429	0.1615	0.1813	0.2023	0.2246	0.2480	0.2984	0.3536	δ
3	1.55	1.57	1.59	1.62	1.65	1.68	1.71	1.74	1.78	1.85	1.94	p
0	20520	19360	18320	17390	16550	15780	15090	14450	13860	12830	11930	Q
9	0.0912	0.1046	0.1189	0.1342	0.1506	0.1679	0.1862	0.2056	0.2259	0.2695	0.3171	δ
1	1.82	1.84	1.87	1.89	1.91	1.94	1.97	2.00	2.03	2.09	2.16	p

spring in inches; b = thickness of originally square bar in inches;
nds for fiber stress of 60000 pounds per square inch in spring section;
coil in inches at load of Q pounds;
hes at no load for fiber stress of 90000 pounds per square inch when compressed solid.

$$\frac{b^3}{4.5\,R + b}; \quad \delta = Q\,\frac{6.2832\,R}{11750000\,b^4}\left[\frac{6}{0.843}R^2 + b^2\right]; \quad R = \frac{D + b}{2}; \quad p = 1.55 + 1.125\,b.$$

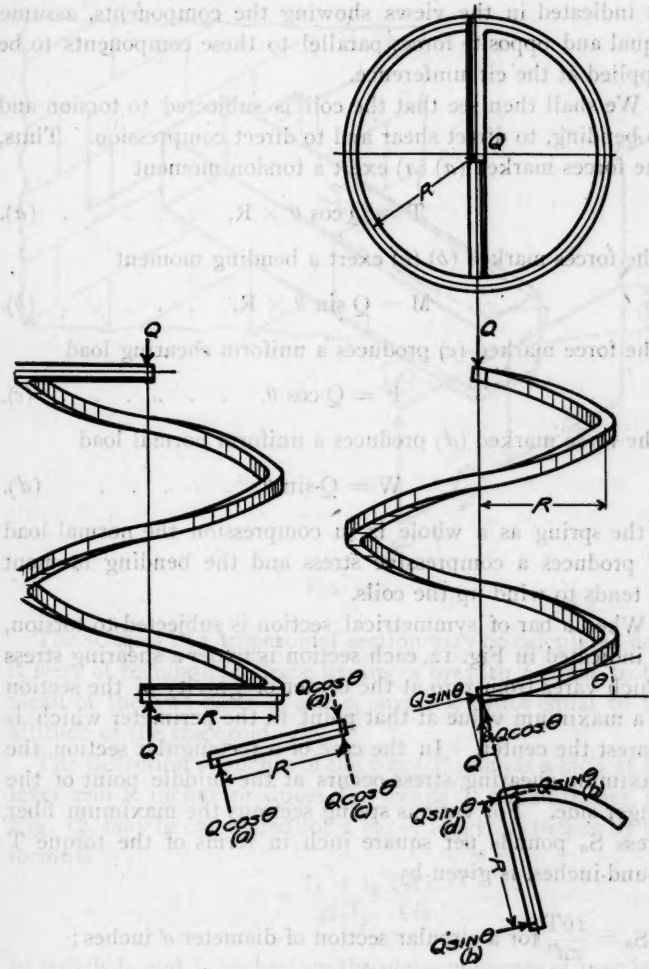


FIG. 11.

the helical spring are represented as bent in at right angles to the coil and extending to the spring axis where the force Q acts. Let Q be resolved into two components parallel and at right angles to the line of inclination of the coils. Also,

as indicated in the views showing the components, assume equal and opposite forces parallel to these components to be applied at the circumference.

We shall then see that the coil is subjected to torsion and to bending, to direct shear and to direct compression. Thus, the forces marked (*a*) (*a*) exert a torsion moment

$$T = Q \cos \theta \times R, \quad \ldots \ldots \quad (a).$$

The forces marked (*b*) (*b*) exert a bending moment

$$M = Q \sin \theta \times R, \quad \ldots \ldots \quad (b).$$

The force marked (*c*) produces a uniform shearing load

$$P = Q \cos \theta, \quad \ldots \ldots \quad (c).$$

The force marked (*d*) produces a uniform normal load

$$W = Q \sin \theta, \quad \ldots \ldots \quad (d).$$

If the spring as a whole is in compression the normal load W produces a compressive stress and the bending moment M tends to wind up the coils.

When a bar of symmetrical section is subjected to torsion, as indicated in Fig. 12, each section is under a shearing stress which vares from zero at the center of gravity of the section to a maximum value at that point in the perimeter which is nearest the center. In the case of a rectangular section the maximum shearing stress occurs at the middle point of the longer side. For various spring sections the maximum fiber stress S_s pounds per square inch in terms of the torque T pound-inches, is given by

$S_s = \frac{16T}{\pi d^3}$ for a circular section of diameter d inches;

$S_s = \frac{16T}{\pi m n^2}$ for an elliptical section m by n inches ($m > n$);

$S_s = \frac{9T}{2b^3}$ for a section b inches square; and

$S_s = \frac{9T}{2bh^2}$ for a rectangular section b by h inches ($b > h$).

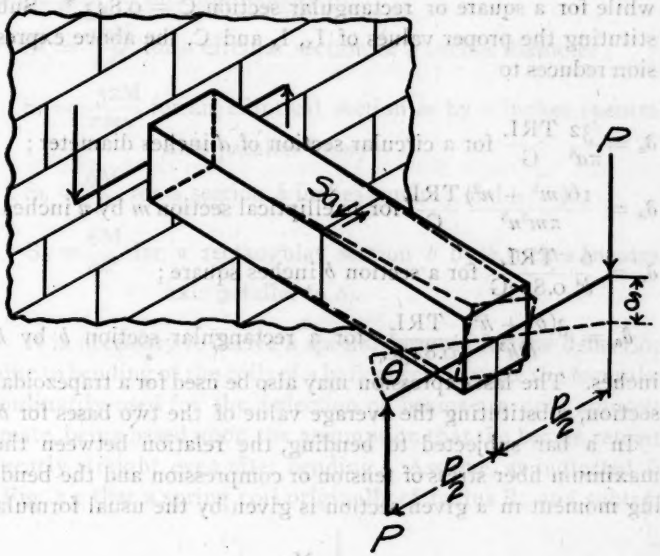


FIG. 12.

The stresses in a trapezoidal section may be calculated as if it were of rectangular shape having a breadth b equal to the mean of the two bases b_1 and b_2 and a width h equal to the altitude of the trapezoid.

The deflection δ_a inches of the force P pounds which at the lever arm R inches produces the torque T pound-inches, see Fig. 12, may be calculated for a bar of length L inches by the formula

$$\delta_a = \frac{I_x + I_y}{4 I_x I_y} \frac{TRL}{CG},$$

in which I_x and I_y inches^4 are the plane moments of inertia of the section of the bar about two axes (the major and the minor) at right angles, G pounds per square inch is the shearing modulus of elasticity of the material used, and C is a constant the value of which corresponds to the shape of the cross-section. Thus, for a circular and an elliptical section C is unity,

while for a square or rectangular section C = 0.843.* Substituting the proper values of I_x, I_y and C, the above expression reduces to

$$\delta_a = \frac{32}{\pi d^4} \frac{TRL}{G}$$ for a circular section of d inches diameter;

$$\delta_a = \frac{16(m^2 + n^2)}{\pi m^3 n^3} \frac{TRL}{G}$$ for an elliptical section m by n inches;

$$\delta_a = \frac{6}{b^4} \frac{TRL}{0.843G}$$ for a section b inches square;

$$\delta_a = \frac{3(b^2 + h^2)}{b^3 h^3} \frac{TRL}{0.843G}$$ for a rectangular section b by h inches. The last expression may also be used for a trapezoidal section, substituting the average value of the two bases for b.

In a bar subjected to bending, the relation between the maximum fiber stress of tension or compression and the bending moment in a given section is given by the usual formula

$$S_b = \frac{Mc}{I},$$

in which S_b = normal stress in pounds per square inch at
c = distance in inches from neutral axis to fiber considered;
M = bending moment in pound-inches;
I = polar moment of inertia in inches^4 of the section.

* The constant 0.843 is for square sections only, but may be applied with but slight error to rectangular sections when the ratio of the sides is 1.5 or even 2, as the following table shows. The values tabulated were derived from a table given by Saint-Venant (see Todhunter and Pearson's "A History of the Elasticity and Strength of Materials," Vol. II, Part I, page 39).

$\frac{b}{h}$	C	$\frac{b}{h}$	C	$\frac{b}{h}$	C
1.0	0.8435	1.8	0.8536	5.0	0.9090
1.1	0.8437	2.0	0.8576	6.0	0.9198
1.2	0.8444	2.5	0.8678	8.0	0.9356
1.3	0.8455	3.0	0.8777	10.0	0.9463
1.4	0.8468	3.5	0.8869	20.0	0.9709
1.5	0.8483	4.0	0.8951	50.0	0.9878
1.6	0.8500	4.5	0.9024	100.0	0.9938

This general formula reduces to

$S_b = \frac{32M}{\pi d^3}$ for a circular section of d inches diameter;

$S_b = \frac{32M}{\pi mn^2}$ for an elliptical section m by n inches (neutral axis parallel to m);

$S_b = \frac{6M}{b^3}$ for a section b inches square; and

$S_b = \frac{6M}{bh^2}$ for a rectangular section b by h inches (neutral axis parallel to b).

It is necessary to derive a special formula for the deflection due to bending of the coils of a helical spring since the formulas ordinarily used for the deflection of beams are only approximate, being based upon the assumption that the beams remain nearly straight even after bending. Assume, as indicated in Fig. 13, that a spring coil originally of radius R_o and subtend

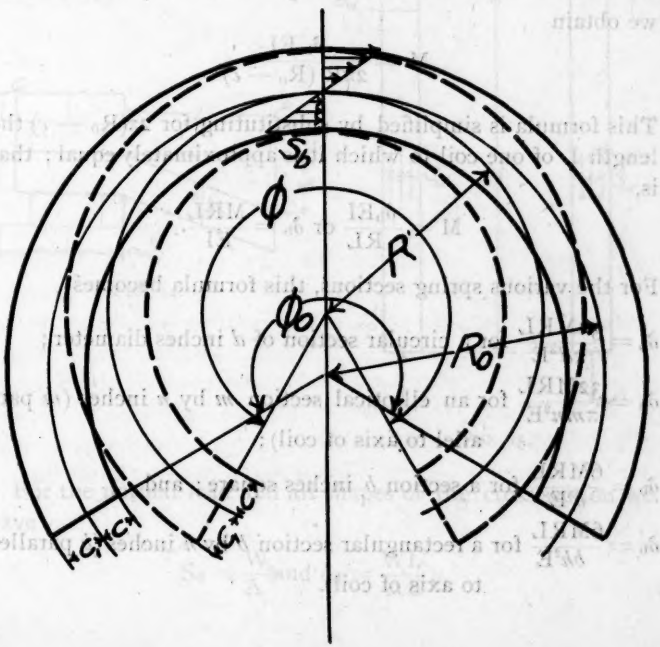


FIG. 13.

ing an angle of φ_o at the center, is strained to a somewhat smaller radius R and a larger angle φ under load, the length of the neutral axis remaining unchanged. We therefore have

$$R_o\varphi_o = R\varphi.$$

Also, the innermost fiber at a distance c from the neutral axis will be subjected to a compressive stress S such that

$$\frac{S}{E} = \frac{(R - c)\varphi - (R_o - c)\varphi_o}{(R_o - c)\varphi_o}$$

where E is the modulus of elasticity of the material. The deflection δ_b of one complete convolution is evidently equal to

$$\delta_b = 2\pi R - 2\pi R_o.$$

Combining these three relations with

$$M = \frac{SI}{c}$$

we obtain

$$M = \frac{\delta_b EI}{2\pi R (R_o - c)}.$$

This formula is simplified by substituting for $2\pi(R_o - c)$ the length L of one coil to which it is approximately equal; that is,

$$M = \frac{\delta_b EI}{RL} \text{ or } \delta_b = \frac{MRL}{EI}.$$

For the various spring sections, this formula becomes

$\delta_b = \frac{32MRL}{\pi d^4 E}$ for a circular section of d inches diameter;

$\delta_b = \frac{32MRL}{\pi mn^2 E}$ for an elliptical section m by n inches (m parallel to axis of coil);

$\delta_b = \frac{6MRL}{b^3 E}$ for a section b inches square; and

$\delta_b = \frac{6MRL}{bh^2 E}$ for a rectangular section b by h inches (b parallel to axis of coil).

For the shearing load and all shapes of the cross-sections we have

$$S_c = \frac{P}{A} \text{ and } \delta_c = \frac{PL}{AG};$$

in which

P = the shearing load in pounds;
A = the area of the section in square inches;
L = the length of the bar in inches;
G = the shearing modulus of elasticity in pounds per square inch;
S_c = the shearing unit stress in pounds per square inch; and
δ_c = the deflection in inches, see Fig. 14.

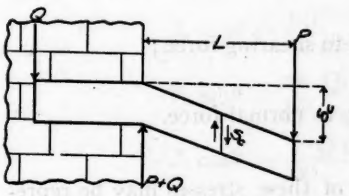


FIG. 14.

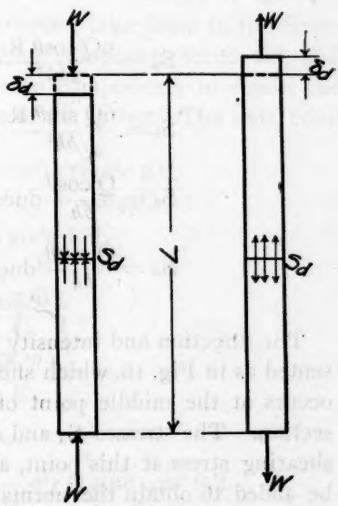


FIG. 15.

For the normal load and all shapes of the cross-section we have

$$S_d = \frac{W}{A} \text{ and } \delta_d = \frac{WL}{AE};$$

in which

W = the normal load in pounds;
A = the area of the section in square inches;
L = the length of the bar in inches;
E = the normal modulus of elasticity in pounds per square inch;
S_d = the normal unit stress in pounds per square inch; and
δ_d = the deflection in inches, see Fig. 15.

Substituting in the above formulas for the maximum fiber stresses in a rectangular (or trapezoidal) bar, the expressions (*a*), (*b*), (*c*) and (*d*) on page 288 for the torsion moment, bending moment, shearing force and normal force in a helical spring, we obtain

$$S_a = \frac{9Q\cos\theta\, R}{2bh^2} \text{ due to torsion;}$$

$$S_b = \frac{6Q\sin\theta\, R}{bh^2} \text{ due to bending;}$$

$$S_c = \frac{Q\cos\theta}{bh} \text{ due to shearing force;}$$

$$S_d = \frac{Q\sin\theta}{bh} \text{ due to normal force.}$$

The direction and intensity of these stresses may be represented as in Fig. 16, which shows that the greatest fiber stress occurs at the middle point of the inside edge of the spring section. The stresses S_a and S_c may be added to obtain the shearing stress at this point, and the stresses S_b and S_d may be added to obtain the normal stress (compression). These sums may then be combined to find the total shearing or normal stress at this same point by one of the following formulas:
Total shearing stress,

$$S_s = \sqrt{(S_a + S_c)^2 + \tfrac{1}{4}(S_b + S_d)^2}.$$

Total normal stress,

$$S_n = \tfrac{1}{2}(S_b + S_d) + \sqrt{(S_a + S_c)^2 + \tfrac{1}{4}(S_b + S_d)^2}.$$

Similarly, we also obtain the several deflections for a spring of rectangular (or trapezoidal) section:

$$\delta_a = \frac{3(b^2 + h^2)}{b^3h^3} \frac{Q \cos\theta \, R^2L}{0.843G} \text{ due to torsion;}$$

$$\delta_b = \frac{12}{bh^3} \frac{Q \sin\theta \, R^2L}{E} \text{ due to bending;}$$

$$\delta_c = \frac{1}{bh} \frac{Q \cos\theta \, L}{G} \text{ due to shear;}$$

$$\delta_d = \frac{1}{bh} \frac{Q \sin\theta \, L}{E} \text{ due to compression or tension.}$$

These several deflections, however, take place in the directions in which the corresponding component forces act, and must accordingly be resolved into components to obtain the axial and tangential deflections of the spring. The axial components are:

$$\delta_a \cos\theta = \frac{3(b^2 + h^2)}{b^3h^3} \frac{Q \cos^2\theta \, R^2L}{0.843G};$$

$$\delta_b \sin\theta = \frac{12}{bh^3} \frac{Q \sin^2\theta \, R^2L}{E};$$

$$\delta_c \cos\theta = \frac{1}{bh} \frac{Q \cos^2\theta \, L}{G};$$

$$\delta_d \sin\theta = \frac{1}{bh} \frac{Q \sin^2\theta \, L}{E}.$$

The tangential components are:

$$-\delta_a \sin\theta = -\frac{3(b^2 + h^2)}{b^3h^3} \frac{Q \sin\theta \cos\theta \, R^2L}{0.843G};$$

$$\delta_b \cos\theta = \frac{12}{bh} \frac{Q \sin\theta \cos\theta \, R^2L}{E};$$

$$-\delta_c \sin\theta = -\frac{1}{bh} \frac{Q \sin\theta \cos\theta \, L}{G};$$

$$\delta_d \cos\theta = \frac{1}{bh} \frac{Q \sin\theta \cos\theta \, L}{E}.$$

The two components marked negative act in the opposite direction to the two positive components.

It is interesting to note that by equating the positive and negative values of the tangential deflections we obtain the expression

$$\frac{3(b^2 + h^2)\,R^2}{b^3h^3 \times 0.843G} + \frac{1}{bhG} = \frac{12R^2}{bh^3E} + \frac{1}{bhE},$$

from which, by assuming $G = \frac{2}{5}E$ and $R = nh$, we obtain

For $n = 3$	$b = 1.74h$,
$n = 4$	$b = 1.72h$,
$n = 5$	$b = 1.71h$,
$n = 6$	$b = 1.70h$,

Springs made of rectangular section and having dimensions in these ratios will give no tangential deflection upon compression or extension.

Now, the question arises as to the relative importance of the four component stresses and the four component axial deflections. This question can best be answered by applying the above formulas to the calculation of the component stresses and axial deflections for the particular spring illustrated at the beginning of this article. For a load of say 6,000 pounds, and assuming G = 11,750,000 pounds per square inch, we have

$S_a = 47{,}210$ pounds per square inch due to torsion;
$S_b = 6{,}040$ pounds per square inch due to bending;
$S_c = 4{,}890$ pounds per square inch due to direct shear;
$S_d = 470$ pounds per square inch due to direct compression.

Adding (*a*) and (*c*), and (*b*) and (*d*), we obtain

$S_a + S_c = 52{,}100$ pounds per square inch shearing fiber stress at the middle point of the inside edge, see Fig. 16;

$S_b + S_d = 6{,}510$ pounds per square inch compression fiber stress at the same point, see Fig. 16.

Combining these stresses by the formulas on pages 294 and 295 we further obtain

$S_s = 52{,}210$ pounds per square inch combined shearing stress;

$S_n = 55{,}465$ pounds per square inch combined compressive stress.

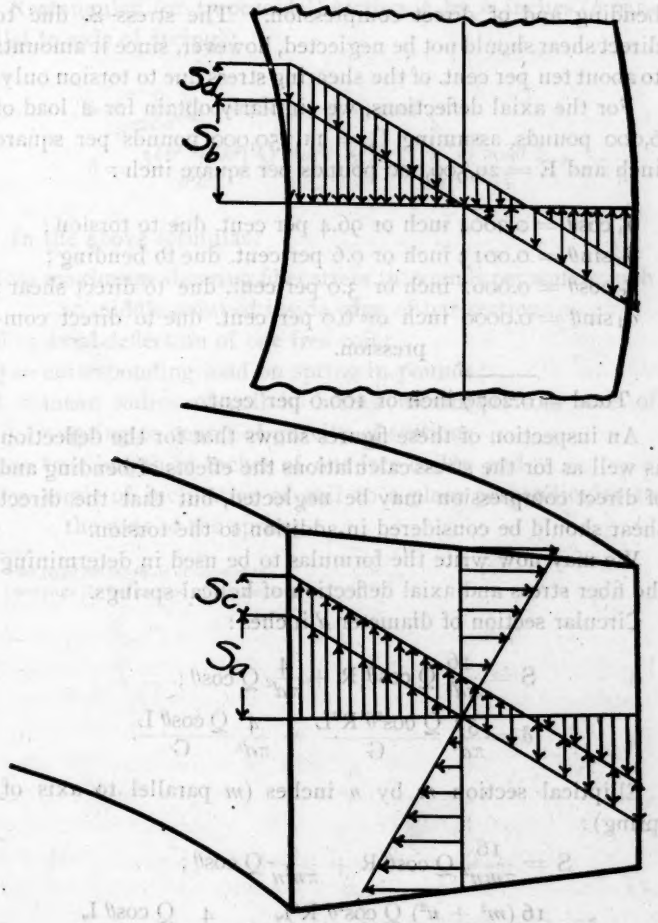


FIG. 16.

Now, the material is much weaker in shear than in compression, so that the above combined shearing stress is a measure of the safety of the spring rather than the slightly higher combined compressive stress. By comparing the combined shearing stress with the sum ($S_a + S_c$) of the shearing stresses due to torsion and to direct shear alone, it is seen that very little error results in neglecting entirely the effects of bending and of direct compression. The stress S_c due to direct shear should not be neglected, however, since it amounts to about ten per cent. of the shearing stress due to torsion only.

For the axial deflections, we similarly obtain for a load of 6,000 pounds, assuming G = 11,750,000 pounds per square inch and E = 29,500,000 pounds per square inch:

$\delta_a \cos\theta$ = 0.2004 inch or 96.4 per cent. due to torsion;
$\delta_b \sin\theta$ = 0.0013 inch or 0.6 per cent. due to bending;
$\delta_c \cos\theta$ = 0.0062 inch or 3.0 per cent. due to direct shear;
$\delta_d \sin\theta$ = 0.0000 inch or 0.0 per cent. due to direct compression.

Total = 0.2079 inch or 100.0 per cent.

An inspection of these figures shows that for the deflection as well as for the stress calculations the effects of bending and of direct compression may be neglected, but that the direct shear should be considered in addition to the torsion.

We may now write the formulas to be used in determining the fiber stress and axial deflection of helical springs.

Circular section of diameter d inches:

$$S = \frac{16}{\pi d^3} Q \cos\theta \, R + \frac{4}{\pi d^2} Q \cos\theta;$$

$$\delta = \frac{32}{\pi d^4} \frac{Q \cos^2\theta \, R^2 L}{G} + \frac{4}{\pi d^2} \frac{Q \cos\theta \, L}{G}.$$

Elliptical section m by n inches (m parallel to axis of spring):

$$S = \frac{16}{\pi m n^2} Q \cos\theta \, R + \frac{4}{\pi m n} Q \cos\theta;$$

$$\delta = \frac{16 (m^2 + n^2)}{\pi m^3 n^3} \frac{Q \cos^2\theta \, R^2 L}{G} + \frac{4}{\pi m n} \frac{Q \cos\theta \, L}{G}.$$

Section b inches square:

$$S = \frac{9}{2b^3} Q \cos\theta R + \frac{1}{b^2} Q \cos\theta;$$

$$\delta = \frac{6}{b^4} \frac{Q \cos^2\theta R^2 L}{0.843G} + \frac{1}{b^2} \frac{Q \cos\theta L}{G}.$$

Rectangular (or trapezoidal) section b by h inches (b parallel to axis of spring):

$$S = \frac{9}{2bh^2} Q \cos\theta R + \frac{1}{bh} Q \cos\theta;$$

$$\delta = \frac{3(b^2 + h^2)}{b^3h^3} \frac{Q \cos^2\theta R^2 L}{0.843G} + \frac{1}{bh} \frac{Q \cos\theta L}{G}.*$$

In the above formulas:

S = maximum shearing fiber stress in pounds per square inch at middle point of inside edge of bar section;

δ = axial deflection of one free coil;

Q = corresponding load on spring in pounds;

R = mean radius of coil in inches measured from axis of spring to center of gravity of section;

L = true length in inches of one free coil†; and

θ = angle of inclination of coil to a place perpendicular to the axis of the spring.

* See page 290 for table of constants.

† See page 276 for method of calculation.

THE PERFORMANCE OF LUBRICATING OIL COOLERS.

By M. C. Stuart, Associate.

The problem of the cooling of lubricating oil is becoming more important, both on account of the necessity of the oil cooler playing its part in the proper operation of the lubricating system, and also on account of the desirability of utilizing in the most economical way the space and weight which may be available for the installation of the cooler. It is the object of this paper to place before those concerned with the operation and design of oil coolers certain features of their performance which have been developed as the result of tests at the U. S. Naval Engineering Experiment Station, Annapolis, Maryland, and to indicate briefly the relation between performance and the weight, volume and amount of cooling surface from comparisons of tests of various types of coolers.

METHOD OF TEST.

An outline of the method of testing oil coolers will serve to lead up to the subject of performance. The cooler to be tested is installed in a manner which duplicates as closely as possible the installation on ship board. Oil is supplied to the cooler from a supply tank by means of a reciprocating or centrifugal pump, and on the way to the cooler passes through an oil heater which serves to bring the oil to any desired temperature. After passing through the cooler, the oil is weighed and returned to the supply tank. A heater is also provided to heat the cooling water to any desired temperature before entering the cooler. The water is also weighed, and the temperatures and pressures of inlet and outlet oil and water are recorded.

With a cooler installed and a certain oil selected for use, the variables for the test are limited to quantities of oil and cooling water circulated, and oil and water temperatures. A test is then divided into two parts; first, runs in which the rates of flow of oil and water are maintained constant, and the temperatures of the inlet oil and water varied, and second, runs in which the inlet temperatures are maintained constant and the rates of flow of oil and water varied.

As an example of the first division of a test, a group of runs is cited in which the quantity of oil circulated through the cooler was 250 pounds per minute and the quantity of cooling water was maintained at 625 pounds per minute. Runs were made with inlet water temperatures of 80 and 90 degrees F. and at each inlet water temperature, runs were made with inlet oil temperatures of 120, 140, 160, and 180 degrees F.

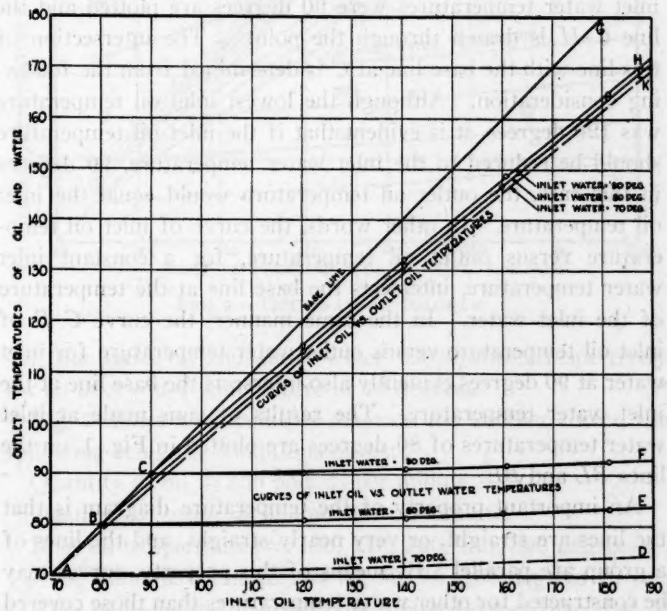


FIG. 1.—TEMPERATURE DIAGRAM.

The resulting outlet temperatures of oil and water are plotted on what is termed a temperature diagram, Fig. 1, which shows graphically the relations between all the temperatures involved in the performance of the cooler at constant rates of flow of oil and water.

TEMPERATURE DIAGRAM.

The temperature diagram has proven to be a valuable aid in analysis of the performance of oil coolers, as well as of other forms of heat transfer apparatus, and it will be discussed in some detail. As shown in Fig. 1, the abscissae are inlet temperatures and the ordinates are outlet temperatures. A base line or equal temperature line, *A–G*, is first drawn diagonally across the diagram. The values of inlet oil temperature versus outlet oil temperature for the runs in which the inlet water temperatures were 90 degrees are plotted and the line *C–H* is drawn through the points. The intersection of this line with the base line at *C* is determined from the following consideration. Although the lowest inlet oil temperature was 120 degrees, it is evident that if the inlet oil temperature should be reduced to the inlet water temperature, 90 degrees in this case, the outlet oil temperature would equal the inlet oil temperature. In other words, the curve of inlet oil temperature versus outlet oil temperature, for a constant inlet water temperature, intersects the base line at the temperature of the inlet water. In the same manner, the curve *C–F* of inlet oil temperature versus outlet water temperature for inlet water at 90 degrees evidently also intersects the base line at the inlet water temperature. The results of runs made at inlet water temperatures of 80 degrees are plotted in Fig. 1, on the lines *BL* and *BE*.

An important property of the temperature diagram is that the lines are straight, or very nearly straight, and the lines of a group are parallel. By means of this property, curves may be constructed for other water temperatures than those covered on a test. Thus, in Fig. 1, the dotted lines *AK* and *AD* repre-

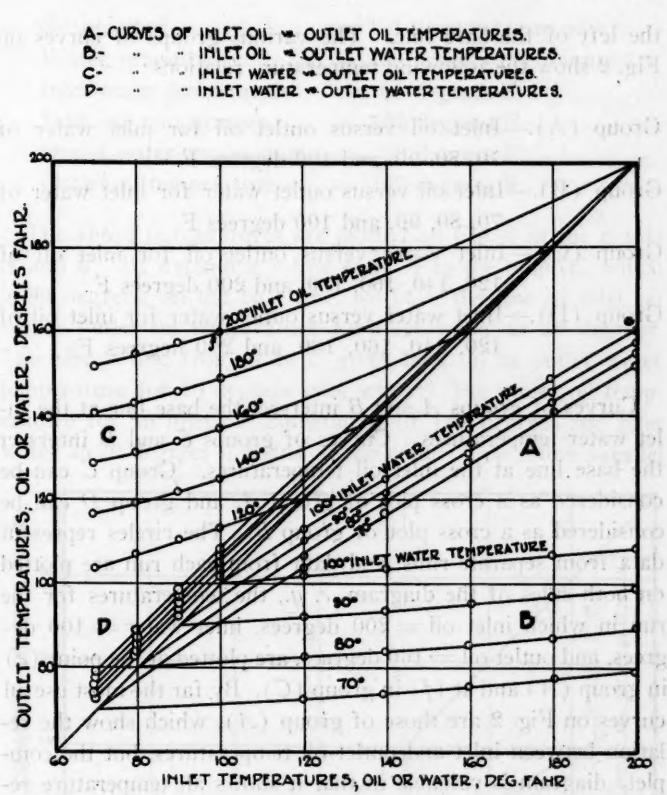


FIG. 2.—TEMPERATURE DIAGRAM.

sent the outlet oil and outlet water temperatures respectively for an inlet water temperature of 70 degrees.

Fig. 2 shows a temperature diagram for another cooler, the following being the conditions of test:

Quantity of oil = 250 pounds per minute.

Quantity of water = 625 pounds per minute.

Inlet oil temperatures = 120, 140, 160, 180, 200 degrees F.

Inlet water temperatures = 70, 80, 90, 100 degrees F.

This temperature diagram goes a step further than that of Fig. 1, in that two additional groups of curves are plotted on

the left of the base line. The various groups of curves on Fig. 2 show the following temperature relations:

Group (A).—Inlet oil versus outlet oil for inlet water of 70, 80, 90, and 100 degrees F.

Group (B).—Inlet oil versus outlet water for inlet water of 70, 80, 90, and 100 degrees F.

Group (C).—Inlet water versus outlet oil for inlet oil of 120, 140, 160, 180, and 200 degrees F.

Group (D).—Inlet water versus outlet water for inlet oil of 120, 140, 160, 180, and 200 degrees F.

Curves of groups *A* and *B* intersect the base line at the inlet water temperatures. Curves of groups *C* and *D* intersect the base line at the inlet oil temperatures. Group *C* can be considered as a cross plot of group *A*, and group *D* can be considered as a cross plot of group *B*. The circles represent data from separate runs, and data from each run are plotted on both sides of the diagram, *e. g.*, the temperatures for the run in which inlet oil = 200 degrees, inlet water = 100 degrees, and outlet oil = 160 degrees, are plotted at the point (*e*) in group (*A*) and at (*f*) in group (*C*). By far the most useful curves on Fig. 2 are those of group (*A*), which show the relation between inlet and outlet oil temperatures, but the complete diagram is valuable in that it shows all temperature relations, and serves as a means of proving the consistency of the temperature data of a group of runs.

Tests made on various coolers and with various rates of flow of oil and water show that in every case the temperature relations may be shown by diagrams similar to those of Figs. 1 and 2. In most cases the lines of the diagrams are straight; in some cases the lines have a slight curvature.

Because of the properties of the temperature diagram, the possibility of the construction of such a diagram from data of a single run suggests itself. The following data are obtained from a test of an oil cooler:

Oil quantity	=	125 pounds per minute.
Water quantity	=	250 pounds per minute.
Inlet water temperature	=	80 degrees F.
Inlet oil temperature	=	160 degrees F.
Outlet water temperature	=	88 degrees F.
Outlet oil temperature	=	127 degrees F.

The above temperatures are plotted on Fig. 3 at the points *A* and *B*. By extending a line from *A* to the point *C*, which is 80 degrees, on the base line, we have the line of inlet oil versus outlet oil temperature for inlet water at 80 degrees. Likewise a line from *B* to *C* gives inlet oil as outlet water temperature for 80 degrees inlet water. The outlet oil temperature for an inlet oil temperature of 140 degrees and inlet water at 70 degrees is found at the point *D*, on a line parallel

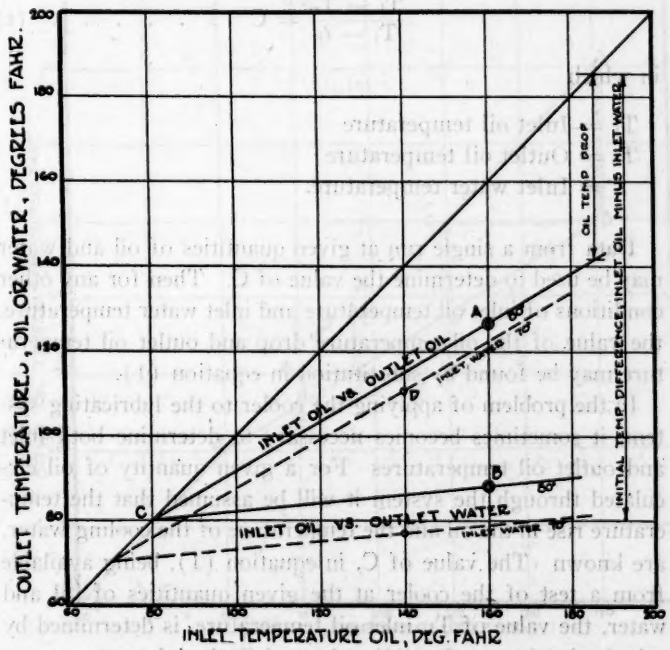


FIG. 3.—CONSTRUCTION FOR TEMPERATURE DIAGRAM.

to AC and through 70 degrees on the base line. Lines for other inlet water temperatures may be drawn in the same manner, and a complete temperature diagram constructed for the given rates of flow of oil and water.

The temperature diagram may be used to derive an analytic method of determining the relation between the various temperatures of the oil cooler. From an inspection of Fig. 3, it is seen that if the lines of the temperature diagram are straight and parallel, there is a constant value of the ratio of the oil temperature drop to the initial temperature difference between the inlet oil and the inlet water. This relation is also evident from the fact that lines of inlet oil versus outlet oil temperature form a constant angle with the base line. Reducing the above principle to a formulae it becomes:

$$\frac{T_i - T_o}{T_i - t_i} = C \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which

T_i = Inlet oil temperature
T_o = Outlet oil temperature
t_i = Inlet water temperature.

Data from a single run at given quantities of oil and water may be used to determine the value of C. Then for any other conditions of inlet oil temperature and inlet water temperature, the value of the oil temperature drop and outlet oil temperature may be found by substitution in equation (1).

In the problem of applying the cooler to the lubricating system, it sometimes becomes necessary to determine both inlet and outlet oil temperatures For a given quantity of oil circulated through the system it will be assumed that the temperature rise in the oil and the temperature of the cooling water, are known The value of C, in equation (1), being available from a test of the cooler at the given quantities of oil and water, the value of T_i, inlet oil temperature, is determined by substitution in equation (1) values of C, the inlet water temp-

erature, and the oil temperature drop in the cooler (which, of course, is the temperature rise in the lubricating system). The outlet oil temperature is found from inlet oil temperature minus temperature drop. This also determines the entering and leaving oil temperatures for the lubricating system. By applying this procedure to various rates of flow of oil, it may be possible to determine the temperatures to be expected in a lubricating system when operating in connection with an oil cooler whose performance is known.

A further use of the temperature diagram in showing the effect of placing two or more coolers in series is shown in Fig. 4. In this figure, *J–B* is the line of inlet oil versus outlet oil temperatures for inlet water of 80 degrees, obtained from a test of a cooler with a certain rate of flow of oil. With inlet

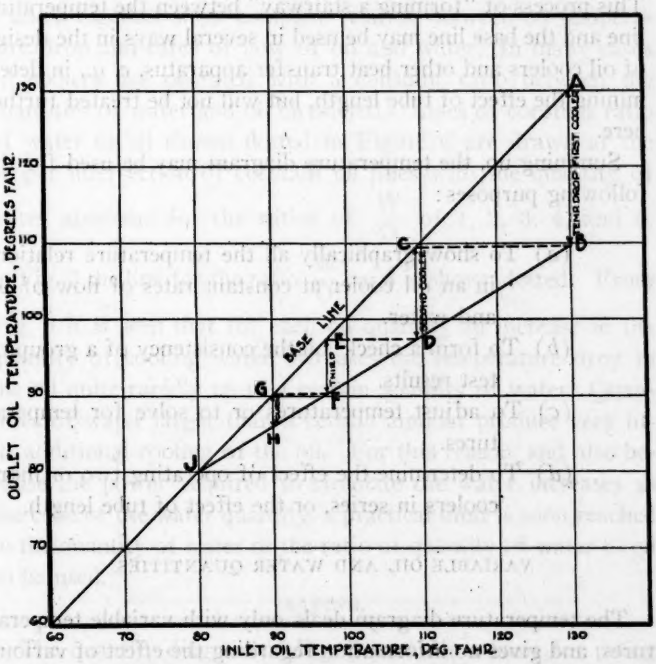


FIG. 4.—USE OF TEMPERATURE DIAGRAM IN DESIGN.

oil to the cooler at a temperature of 130 degrees, therefore, the outlet oil temperature will be at the point *B* or 109.5 degrees. A second cooler is placed in series and supplied with water at 80 degrees. The outlet oil temperature of the first cooler then becomes the inlet oil temperature of the second cooler. The outlet temperature *B* of the first cooler is transferred to the inlet temperature of the second cooler by the simple process of projecting the point *B* horizontally to the point *C* on the base line. If the second cooler is similar in construction to the first, and supplied with the same amount of cooling water at 80 degrees, the oil entering at a temperature of 109.5 degrees will have an outlet temperature as determined at *D*, or 97.5 degrees. The outlet temperatures of the 3d and 4th coolers in series will be at *F* and *H* respectively. This process of "forming a stairway" between the temperature line and the base line may be used in several ways in the design of oil coolers and other heat transfer apparatus, *e. g.*, in determining the effect of tube length, but will not be treated further here.

Summing up, the temperature diagram may be used for the following purposes:

(*a*) To show graphically all the temperature relations in an oil cooler at constant rates of flow of oil and water.

(*b*) To form a check on the consistency of a group of test results.

(*c*) To adjust temperatures or to solve for temperatures.

(*d*) To determine the effect of operating two or more coolers in series, or the effect of tube length.

VARIABLE OIL AND WATER QUANTITIES.

The temperature diagram deals only with variable temperatures, and gives no information regarding the effect of various rates of flow of oil or water. The second division of a test of

an oil cooler consists of maintaining constant temperatures of inlet oil and inlet water and varying the rates of flow of oil and water. The results of such a group of runs are plotted in Fig. 5, in what is termed a "quantity diagram." All of the runs of Fig. 5 were made with inlet water temperatures of 80 degrees and inlet oil temperature of 130 degrees. Runs were made at rates of flow of water equal to 100, 200, 500, and 800 pounds per minute, and at each of these rates of flow of water runs were made with rates of flow of oil from 100 to 450 pounds per minute. The outlet oil temperatures for each run are plotted against oil quantities and curves drawn for equal water quantities. The curves of Fig. 5 are cross plotted in Fig. 6 to coordinates of water quantity and outlet oil temperature with curves of equal oil quantities. Figs 5 and 6 taken together show complete relation between oil temperature drop and rates of flow of oil and water. In many cases oil coolers are operated with a constant ratio between the quantities of water and oil circulated. Lines of constant ratio of water to oil shown dotted in Figure 6 are drawn at the proper intersection of constant oil lines with the quantity of water abscissas for the ratios of $\frac{W}{O}$ of 1, 2, 3, 4, and 5. In Fig. 5 the line for the ratio $\frac{W}{O} = 3$ is shown dotted. From Fig. 6 it is seen that for each oil quantity an increase in the quantity of cooling water increases the temperature drop in the oil quite rapidly up to a certain quantity of water. Quantities of water larger than a certain amount produce very little additional cooling of the oil. For this reason, and also because the power required to circulate the water increases as the cube of the water quantity, a practical limit is soon reached to the quantity of water or the ratio of quantity of water to oil to be used.

RATING.

The quantity diagram of Fig. 5 provides a basis for rating or determining the capacity of an oil cooler. To say that a

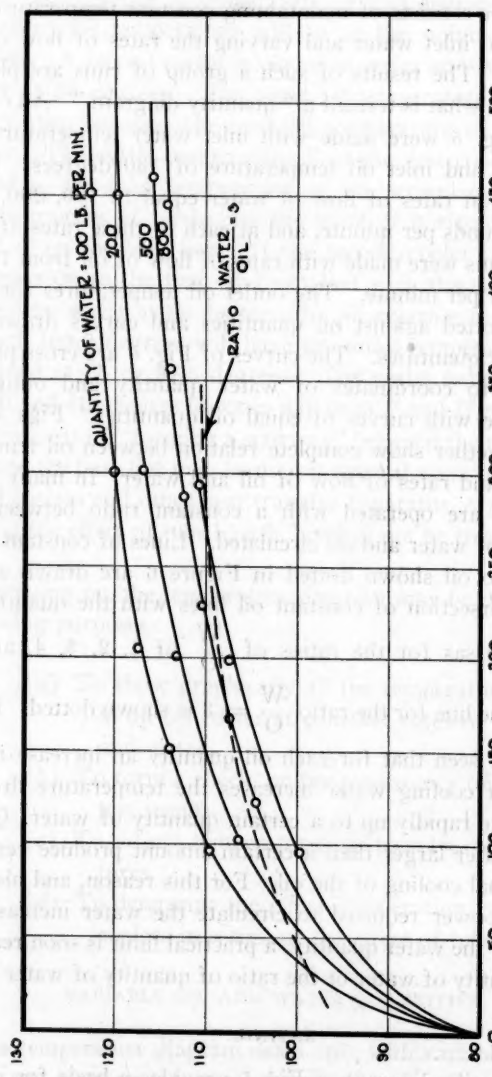


FIG. 5.—QUANTITY DIAGRAM.

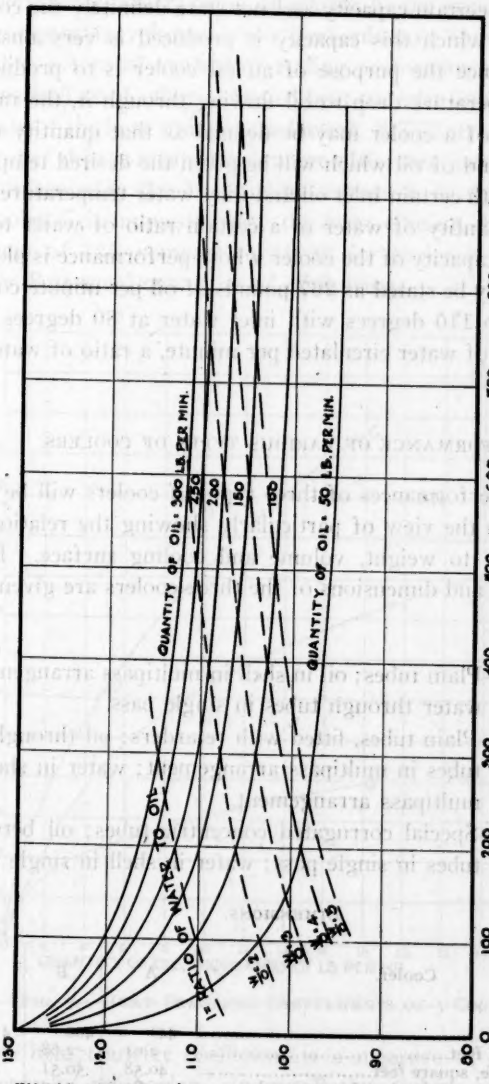


FIG. 6.—QUANTITY DIAGRAM.

cooler has a certain capacity and not state definitely the conditions under which this capacity is produced is very unsatisfactory. Since the purpose of an oil cooler is to produce a certain temperature drop to oil flowing through it, the rating or capacity of a cooler may be defined as that quantity of a particular kind of oil which will be given the desired temperature drop with certain inlet oil and inlet water temperature and a certain quantity of water or a certain ratio of water to oil used. The capacity of the cooler whose performance is plotted in Fig. 5 may be stated as 267 pounds of oil per minute cooled from 130 to 110 degrees with inlet water at 80 degrees and 800 pounds of water circulated per minute, a ratio of water to oil of 3.

PERFORMANCE OF VARIOUS TYPES OF COOLERS.

Typical performances of three types of coolers will be presented, with the view of particularly showing the relation of performance to weight, volume and cooling surface. Brief descriptions and dimensions of the three coolers are given following:

Cooler A:—Plain tubes; oil in shell in multipass arrangement; water through tubes in single pass.

Cooler B:—Plain tubes, fitted with retarders; oil through the tubes in multipass arrangement; water in shell in multipass arrangement.

Cooler C:—Special corrugated concentric tubes; oil between tubes in single pass; water in shell in single pass.

DIMENSIONS.

Cooler.	A	B	C
Weight, pound	441	400	416
Volume, cubic feet	2.01	2.68	1.64
Cooling surface, square feet	40.58	50.51	10.67
Weight per square foot of surface, pounds	10.87	7.95	39.00
Volume per square foot of surface, cubic foot	.049	.053	.15

These three coolers being of practically the same weight and of comparable volumes, their performances should be comparable. A usual method of stating performance of oil coolers, as well as of other forms of heat transfer apparatus, is by the use of the heat transfer coefficient, B.T.U. transferred per hour, per degree mean temperature difference, per square foot of surface. On this basis of performance, as shown in Fig. 7, cooler *C* is unquestionably the most efficient, having a coefficient of 125, compared with 82 for cooler *A* and 39 for cooler *B* at equal quantities of oil flowing.

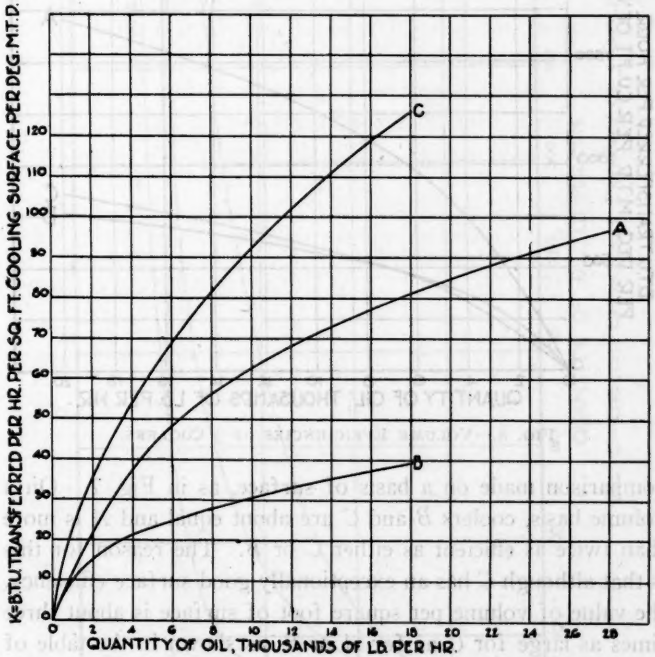


FIG. 7.—HEAT TRANSFER COEFFICIENTS OF 3 COOLERS.

The heat transfer coefficient is a measure of the surface efficiency of the cooler. In installations where space is at a premium the efficiency of the volume of the cooler is far more

important than surface efficiency. In this instance each unit of volume occupied must be used to transfer as large an amount of heat as possible. A factor which places performance in a basis of volume of the apparatus is B.T.U. transferred per hour, per degree mean temperature difference, per cubic foot of volume. A comparison of the coolers upon this basis, as plotted in Fig. 8, shows an entirely different result from the

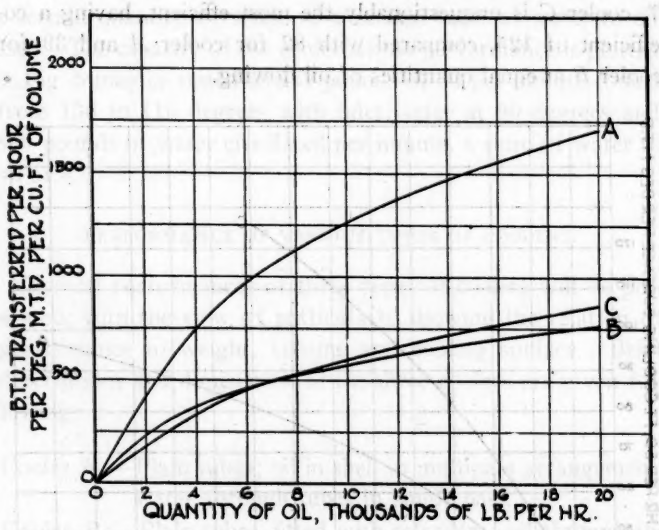


FIG. 8.—VOLUME EFFICIENCIES OF 3 COOLERS.

comparison made on a basis of surface, as in Fig. 7. On a volume basis, coolers *B* and *C* are about equal and *A* is more than twice as efficient as either *C* or *B*. The reason for this is that although *C* has an exceptionally good surface efficiency, the value of volume per square foot of surface is about three times as large for *C* as for *A* or *B*, as shown in the table of dimensions. The cooling surface of *C*, though efficient in itself, is of such a nature that it is impossible to get a large amount of it into a small space. A more direct method of stating and comparing performances than either of the preceding is one on a basis of actual amount of cooling given to

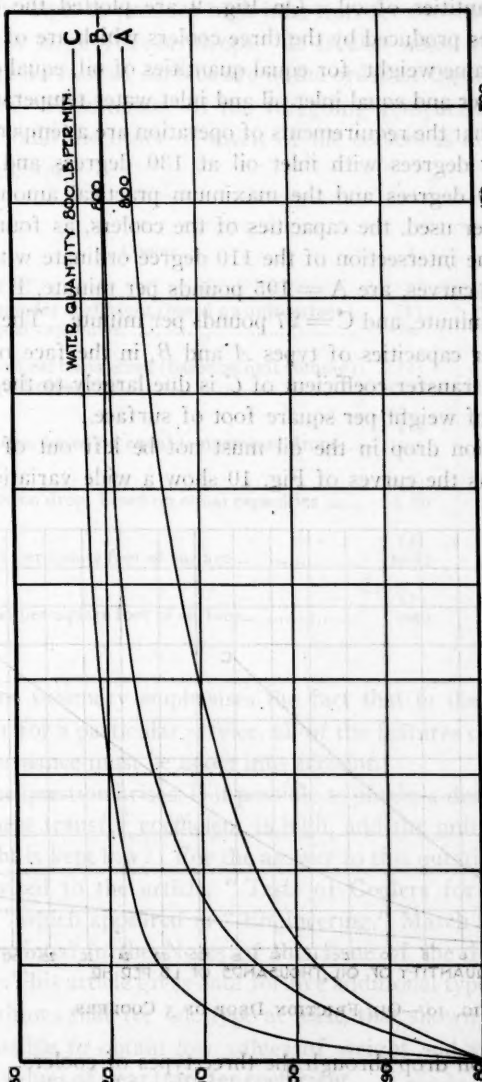


FIG. 9.—COOLING EFFECT AND CAPACITY OF 3 COOLERS.

various quantities of oil. On Fig. 9 are plotted the outlet temperatures produced by the three coolers which are of practically the same weight, for equal quantities of oil, equal quantities of water and equal inlet oil and inlet water temperatures. Assuming that the requirements of operation are a temperature drop of 20 degrees with inlet oil at 130 degrees and inlet water at 80 degrees and the maximum practical amount of cooling water used, the capacities of the coolers, as found on Fig. 9, at the intersection of the 110 degree ordinate with the temperature curves, are A = 195 pounds per minute, B = 85 pounds per minute, and C = 27 pounds per minute. The relatively higher capacities of types *A* and *B*, in the face of the higher heat transfer coefficient of *C* is due largely to the relative values of weight per square foot of surface.

The friction drop in the oil must not be left out of consideration, as the curves of Fig. 10 show a wide variation in

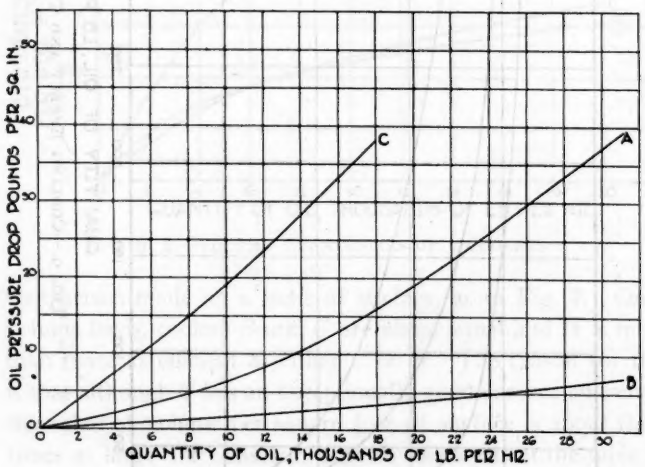


FIG. 10.—OIL FRICTION DROP OF 3 COOLERS.

the oil friction drop through the three types of coolers. It is of interest to note that the relative friction drops are in the same order as the relative heat transfer factors, indicating

that attempts to produce high heat transfer coefficients usually produce large friction drops. It is also noted that weights per square foot of surface are in the order of the heat transfer factors. A summary of the foregoing comparisons is given below, and the order of merit of the coolers is noted in each comparison.

Cooler.	A	B	C
Heat transfer coefficient (based on unit surface) at equal capacities	(2) 82	(3) 39	(1) 125
Relative heat transferred (based on unit volume) at equal capacities	(1) 1,000	(2) 450	(2) 450
Capacities, based on equal temperature drops	(1) 195	(2) 85	(3) 27
Oil-friction drop, based on equal capacities	(2) 7.80	(1) 2.00	(3) 22.6
Weight per square foot of surface	(2) 10.87	(1) 7.95	(3) 39.00
Volume per square foot of surface	(1) .049	(2) .053	(3) .15

This summary emphasises the fact that in the choice of a cooler for a particular service, all of the features of design and performance must be taken into account.

The question arises, is it possible to obtain a design in which the heat transfer coefficient is high, and the unit volume and weight is kept low? For the answer to this question, attention is invited to the article, "Tests of Coolers for Lubricating Oils," which appeared in "Engineering," March 2, 1917, and is reprinted in the Notes of this issue of the Journal, page 339. This article gives data for five additional types of coolers, and shows that for one type at least, that shown in Fig. 8, it is possible to obtain low values of weight and volume, with high values of heat transfer coefficient.

Summing up the subject of relative performance and merit

of various types of coolers, the statement may be made that that cooler is best which will produce a given temperature drop to a given quantity of oil with the least volume and weight, and without excessive friction drop. Any feature of design which produces a high heat transfer coefficient will decrease the amount of surface required to accomplish a desired result, but a high heat transfer coefficient is of no value unless the reduction in surface is accompanied by a proportionate reduction in volume and weight, and is not accompanied by excessive friction losses and impractical mechanical features.

THE MAIN ENGINES OF THE U. S. S. *CUYAMA*.

By LIEUT. COMMANDER F. J. CLEARY, U. S. N., MEMBER.

The Mare Island method of handling the main engines of the U. S. S. *Cuyama* is an example of what may be accomplished by skilful handling of apparently inadequate equipment.

Each main engine was lifted from the erecting floor of the machine shop, landed on a flat car, hauled to the water front and hoisted aboard ship as an assembled unit. As the main engines of the U. S. S. *Kanawha* of the identical type and weight had been similarly handled in 1914, no difficulty was anticipated.

Each engine, fully assembled, weighed about 103 tons, a weight materially in excess of the capacity of the heaviest equipment at Mare Island, which was as follows:

100-ton shears on water front;
40-ton traveling crane in the machine shop;
50-ton flat car.

The engines were completely erected in the shop as usual. While on the erecting floor two lines were run fore and aft on the extreme outboard and inboard sides of the engine column seatings. These lines were supported by brackets secured to the columns, rested to ball-bearing sheaves and were kept taut by 75-pound weights on each end. Micrometer readings were taken on all column seatings and these readings were used when fitting the chocks in the engine rooms.

The following details which could be disassembled without affecting the alignment of the moving parts, were removed:

Cylinder covers,
Receiver piping,
Oiling system,
Water service,
Jacking engine,
Cylinder lagging plates,
Relief valves and drain cocks,

reducing the weight of each engine to 92.7 tons.

A few days before it was desired to move the first (port) engine, the 50-ton flat car was loaded with 115 tons of chain and hauled over the track from the machine shop to the shears. The track was carefully watched during this trip and the ballast tamped down wherever the road bed appeared soft. The shears were tested by hoisting, racking out and lowering 130 tons of anchors.

The 40-ton crane in the shop was used to lift the H.P. end of the engine high enough to slide a number of well-tallowed 1-inch × 6-inch flat steel bars under the engine bedplate.

A heavy purchase was hooked to the engine, the fall led to a 4-foot diameter casting bolted to the table of a 14-foot Niles Vertical Boring Mill with 20-H.P. motor, and the engine hauled out on the car track at the end of the shop.

The H.P. end of the engine was lifted by the 40-ton crane and a heavy square timber placed under the center of the engine bedplate, the engine tilted on this timber and one end blocked up. By alternately lifting the ends of the engine and by increasing the height of the central timber, the engine was gradually blocked up high enough so that the flat car could be run under it. By a reverse method blocking was removed until the engine rested on the flat car, the entire operation from the time of starting the engine from the erecting floor to landing it on the car taking about 12 hours.

The flat car was reinforced by 16-inch × 16-inch timbers lashed longitudinally along each side of the car platform. A 12-inch × 12-inch timber was lashed across under the center of the 16-inch × 16-inch timbers. Under each end of each 16-inch × 16-inch was lashed a 12-inch × 12-inch block, un-

derneath that a 4-inch × 12-inch block and below that a 12-inch × 12-inch timber extending across the end of the car. All this blocking was secured by being lashed together and to the engine columns.

The port engine left the shop at 12:45 P. M., Nov. 22, 1916, and reached the shears at 1:10 P. M., began hoisting at 4:00 P. M. and was landed in the engine room at 6:40 P. M.

The engine was lifted by four wire pendants, two 52 feet and two 25 feet in length. These pendants were 5-inch wire. The long pendants had 18-inch eyes spliced in the ends. The short pendants had a shackle at one end and an 18-inch eye at the other end. One long pendant was used on each end of the engine; the pendant was rove through the large shackle on the shear blocks, the eyes coming down on opposite sides of the engine. The eyes were then passed through the crotch of the end columns and secured by passing a 3½-inch steel toggle through each eye. The short pendants were used on the center columns, one on each side of the engine. The pendant shackle was lashed to the shackle on the shear blocks, the eye being passed through the crotch of the center column and secured by passing a 3½-inch steel toggle through the eye. Wooden blocks 6 inches × 12 inches × 3 feet long were lashed on the inside of the columns for the toggles to bear against. Chafing battens were secured wherever necessary and tackles were used to swing the engine as required.

There were several days of heavy rain immediately after the port engine was placed aboard. On December 4th a locomotive crane, loaded to 110 tons, was run over the track from the machine shop to the shears to give assurance that the rain had not softened the road bed.

The starboard engine was handled by the same method; it was placed on the flat car on December 4th. It left the machine shop at 8:45 A. M., December 5th, reached the shears at 9:00 A. M., began hoisting at 2:00 P. M., and was landed in the engine room at 4:30 P. M.

The engines were landed in the engine room on 1-inch × 6-inch × 12-foot steel bars, well greased, and jacked into

position by four 15-ton hydraulic jacks. The lines were again run over the column seatings. The engines were then wedged into place, both vertical and horizontal alignments being made by the shaft-coupling flanges. Six chocks were first fitted, three on each side. These chocks had body-bound bolts and were set up tight, reference being made to the micrometer readings. Another set of chocks was then fitted between the first set. As each new group of chocks was fitted, a method of checking back was used by removing the bolts of the chocks previously fitted. If any chocks could be moved, it was apparent that one or two adjacent chocks were too thick. These chocks were then refitted, always using, as reference, the two lines above the engine.

After all the chocks were fitted, the holding-down bolts set home and the coupling bolts in place, the alignment was so accurate that both engines could be jacked with a single jacking engine. One hundred and fifty-nine chocks were fitted under the port engine, 150 chocks under the starboard engine; 20 of the chocks under each engine had body-bound bolts.

No parts of the engines were disturbed by transportation, nor were any readjustments of bearings found necessary after the installation of the engines in the ship.

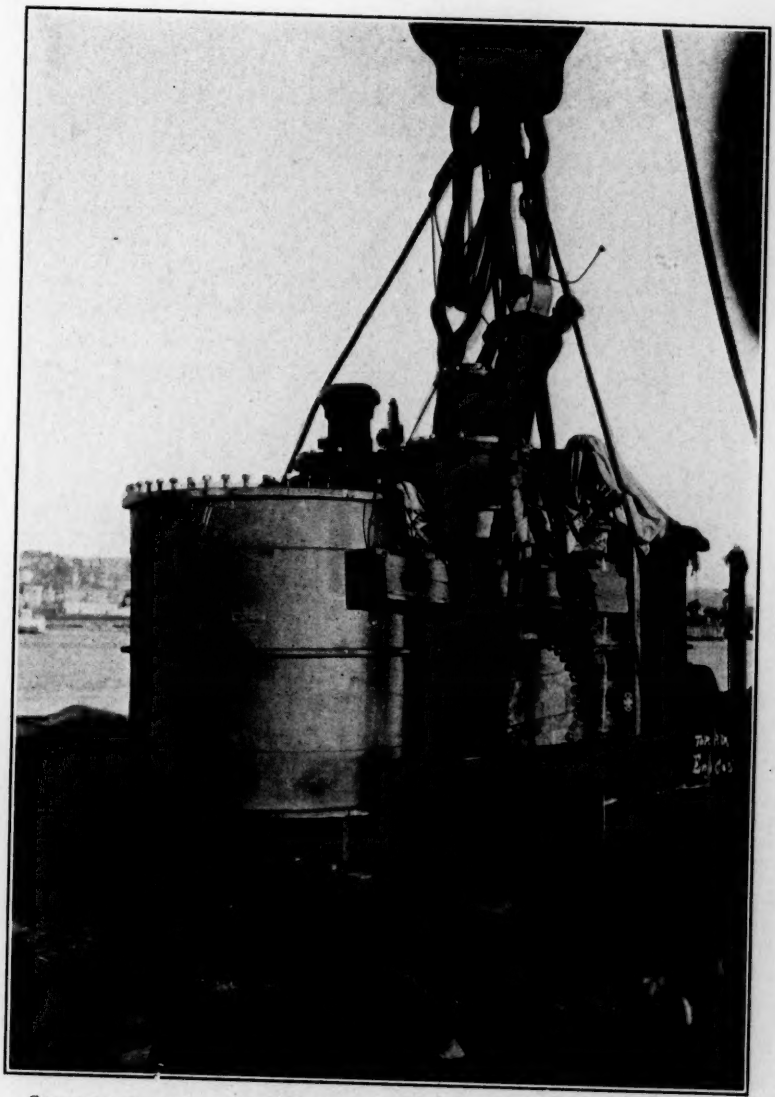
It is estimated that this method of placing these engines aboard ship assembled, saved about 60 days' time and between $8,000 and $10,000 in labor.

The U. S. S. *Cuyama* is designed for carrying fuel oil, etc., and is the third ship of this type to be built at Mare Island.

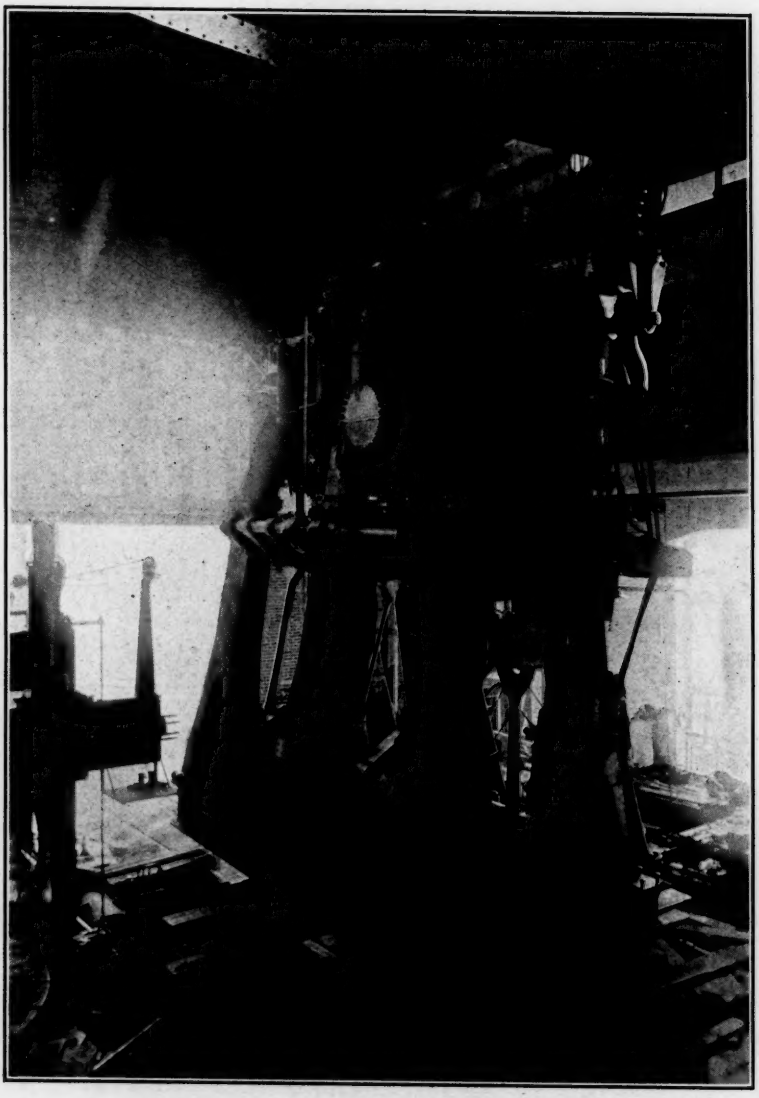
The *Kanawha* was built and engined at Mare Island and has been giving excellent service for about two years.

The *Maumee* was built at Mare Island and towed to New York for the installation of the Diesel engines built at the New York Navy Yard.

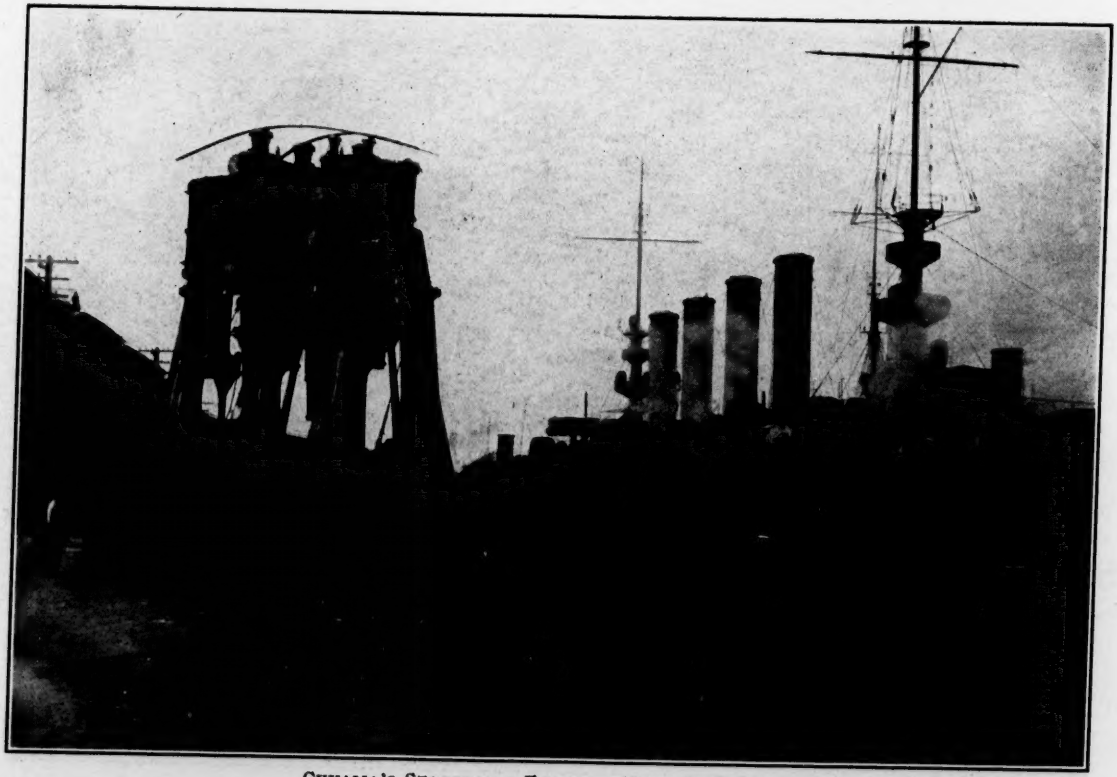
The *Cuyama* is a sister ship of the *Kanawha* in all respects except as to the location of the bridge and various changes in deck fittings and cargo tank arrangement. Her main characteristics are as follows:



CUYAMA'S STARBOARD ENGINE. PASSING THROUGH ENGINE ROOM HATCH.



CUYAMA'S STARBOARD ENGINE. LIFTING AND BLOCKING UP.



CUYAMA'S STARBOARD ENGINE. NEAR SEA WALL.

Mean trial displacement, tons,	14,500
Full load displacement, tons,	15,100
Draught at mean trial displacement, ft. and ins.,	26–4
Draught at full load displacement, feet and ins.,	27–4

Steam is supplied by four Babcock & Wilcox oil-burning water-tube boilers, each boiler having a single furnace or combustion chamber fitted with seven Peabody burners.

Total heating surface of the four boilers, square feet	12,240
Total furnace volume of the four boilers, cubic feet	1,312
Ratio cubic feet furnace volume to square feet H.S. =	1 to 9.33
Gage steam pressure at boilers, pounds	200
Gage steam pressure at H.P. steam chest, pounds	190

The main engines are right and left-handed, 3-cylinder, vertical, inverted, direct-acting, triple-expansion, and are fitted with the Lovekin inertia cylinders.

All castings for these engines were cast in the navy-yard foundry.

The principal characteristics and dimensions are:

Height over all, feet and inches,	25–5½
Length of bedplate, feet and inches,	20–7⅛
Width of bedplate, feet and inches,	11–1
Diameter of crank pins, inches,	13
Diameter of main bearings, inches,	13
Diameter of piston rods, inches	5½

All frames are cast iron inverted "Y's."

Diameter of cylinders:

H.P., inches,	23
I.P., inches,	39½
L.P., inches,	68½
Stroke, inches,	48

I.H.P. both main engines (100 revolutions per minute), 5,200.

The cranks are at 120 degrees, the sequence being high pressure, intermediate pressure, low pressure.

The main steam valves are operated by the Stephenson link motion with double-bar links.

There are:

One H.P. piston valve, diameter, inches,	12
Two I.P. piston valves, diameter, inches,	12
Two L.P. piston valves, diameter, inches,	$23\frac{1}{2}$

The H.P. valves are inside valves. The I.P. and L.P. valves are outside valves.

H.P. valves are, length, feet and inches,	$5-5\frac{13}{16}$
I.P. valves are, length, feet and inches,	$5-5\frac{13}{32}$
L.P. valves are, length, feet and inches,	5–8

The valve travel is:

H.P., inches,	$7\frac{3}{4}$
I.P., inches,	$7\frac{3}{4}$
L.P., inches,	$9\frac{1}{2}$

The angular advance is:

H.P.,	41° 24′
I.P.,	43° 18′
L.P.,	47° 27′

The designed cut-off in decimals of stroke is:

	Top.	Bottom.
H.P.,	.693	.617
I.P.,	.700	.620
L.P.,	.641	.564

The valves can be linked into cut-off at half stroke on all cylinders.

EXAMINATION OF OILS FROM THE ATLANTIC FLEET, BEFORE AND AFTER USE.

By J. G. O'NEILL, CHEMIST, ASSOCIATE.

The object of this test was to examine chemically force-feed lubricating oils, before and after use, to determine the amount and nature of the changes which force-feed lubricating oils undergo in service; and to designate the simplest manner in which to treat used oils in order to make them of as good quality as before use.

OILS USED IN TEST.

In Table I are given the names and the classification of the used and unused oils tested.

TABLE I.

Name of oil.	Classification:
Used "A,"	Heavy force feed.
Unused "A,"	Heavy force feed.
Used "B,"	Medium force feed.
Unused "B,"	Medium force feed.
Used "C,"	Medium force feed.
Unused "C,"	Medium force feed.
Used "D,"	Medium force feed.
Unused "D,"	Medium force feed.
Used "E,"	Light force feed.
Unused "E,"	Light force feed.
Used "F,"	Light force feed.
Unused "F,"	Light force feed.

In Table II are given the number of hours which the oil had been in use, the percentage of new oil added during this time and the maximum temperature at the bearings.

TABLE II.

Name of used oil.	No. of hrs. in use.	Percentage of new oil added.	Maximum temp. at bearings.
"A"	540	27.3	124 F.
"B"	865	14.30	118 F.
"C"	540	21.8	122 F.
"D"	1,300	23.63	110–122 F.
"E"	1,046*	38.80	118 F.
"F"	675	10.52	118 F.

METHODS OF TESTING AND RESULTS.

Many of the methods used in this test were either special chemical methods taken from technical books on the testing of lubricating oils, or were methods recently developed at the Engineering Experiment Station. A large number of the ordinary routine chemical tests for lubricating oils were found to be valueless for the purposes of this test. The new, or special chemical tests used will be explained in detail.

TOTAL SULPHUR.

(*a*) Information was desired in regard to any changes which might take place in the total sulphur content of an oil during use. For determining total sulphur in the oils the Conradson method was used. This method consists of burning in a small kerosene lamp, fitted with a wick, a 5-gram sample of the oil diluted with a known amount of kerosene in which the sulphur has been carefully determined. The mixture of lubricating oil and kerosene was drawn by capillarity through the wick and burned, the lamp being so fitted that the products of combustion were drawn through a tower down which trickled a weak standard solution of sodium carbonate. When the oil was completely burned the tower was thoroughly washed with distilled water. Combining the solution of sodium carbonate from the tower and the washings, it was quickly determined

*200 gallons of old oil boiled out at 325 F. had been added during this period to oil "E."

by means of a standard hydrochloric-acid solution how much of the sodium carbonate solution had become neutralized by the sulphur dioxid in the vapors from the lamp. Knowing the quantity of sulphur in the kerosene used to dilute the lubricating oil, the sulphur in the lubricating oil was easily calculated. In Table III are given the results obtained.

TABLE III.

Percentage of Sulphur.

Name of oil.	Original oil.	Used oil.
A	0.17	0.17
B	0.20	0.16
C	0.06	0.06
D	0.15	0.16
E	0.05	0.06
F	0.07	0.08

Organic Acidity.

(*b*) Organic acidity was determined according to Holde's method in which 20 cc. of oil are shaken in a glass-stoppered measuring flask with 40 cc. of neutralized alcohol. After standing over night, one-half of the alcohol layer is poured off, diluted with neutralized 96 per cent. alcohol and titrated with N/10 alcoholic sodium hydroxide in the presence of Alkali Blue 6 b. For purposes of comparison the original oils were oxidized by the General Electric Company's oxidation method, known widely as the G. E. test, in which 100 grams of the oil in an Erlenmeyer flask were heated in an oil bath to 120 degrees C. for 72 hours. During this time a small stream of oxygen was bubbled through the oil. In table IV are given the organic acidity of the original oil, of the used oil and of the original oil treated according to the G. E. test. Acidity is expressed as acid number, which means the number of milligrams of caustic potash required to neutralize one gram of the oil.

TABLE IV.

Acid Number.

Name of oil.	Original oil.	Used oil.	Original oil after G. E. test.
A	.30	.43	.76
B	.15	.56	.42
C	.06	.19	.20
D	.06	.13	.31
E	.06	.93	.25
F	.06	.19	.31

Asphaltenes.

(*c*) The asphaltenes are the black particles apparent in oil which has been used for some time. They are one of the constituents of asphalt, and are formed from the condensation of the resins in a lubricating oil. They are determined by dissolving 10 grams of the oil in 50 cc. of 88-degree gasoline, allowing to stand over night, and filtering on a weighed Gooch crucible prepared with an asbestos pad. The Gooch crucible is washed thoroughly with 88-degree gasoline, dried at 105 degrees C., and the increase in weight calculated as asphaltenes. Information was desired as to the amount of asphaltenes which would be formed by oxidizing an oil. Some of the original oils were oxidized according to the G. E. test (sub-paragraph *b*), and all of the original oils were oxidized according to the Waters' carbonization test, and the asphaltenes determined. The Waters' carbonization test, devised by Dr. C. E. Waters of the Bureau of Standards, was carried out in a crude apparatus constructed at this Station; and it consists of heating 10 grams of the oil in a 125-cc. Erlenmeyer flask in an air bath for 2½ hours at 250 centigrade. Time is counted from the minute when the heat is applied, the flasks being left open to the air. In table V are given the asphaltenes in the original oil, in the used oil, in a few of the original oils after the G. E. test and in the original oils after Dr. Waters' test.

TABLE V.

Percentage by Weight of Asphaltenes.

Name of oil.	Original oil.	Used oil.	Original oil after G. E. test.	Original oil after Waters' test.
A	None.	.083	. . .	.024
B	None.	.043	. . .	.050
C	None.	.015	.024	.013
D	None.	.031	.042	.022
E	None.	.015	.054	.006
F	None.	.030	. . .	.056

Saponification Number.

(*d*) Saponification resins are formed by the oxidation of lubricating oils; and in badly oxidized oils, or in poorly refined oils, these resins are present in considerable quantity. To determine the saponification number of an oil, a weighed quantity of the oil, with a known amount of standard alcohol caustic potash, is boiled in a flask with a reflux condenser. The contents of the flask are then cooled and the excess caustic potash is determined by means of a standard hydrochloric acid solution. Knowing the quantities of oil, standard alkali and standard acid used, the number of milligrams of caustic potash required to saponify one gram of the oil is calculated, after subtracting the number of milligrams of caustic potash required to neutralize the organic acidity in one gram of the oil. This is called the saponification number. In Table VI are given the saponification numbers of the original oils, the used oils, some of the original oils which have been subjected to the G. E. test, and the original oils subjected to the Waters' test.

TABLE VI.

Saponification Number.

Name of oil.	Original oil.	Used oil.	Original oil after G. E. test.	Original oil after Waters' test.
A	2.50	3.90	. . .	3.98
B	1.13	5.64	. . .	4.68
C	1.11	2.95	2.84	3.98
D	1.36	2.67	2.49	3.04
E	2.63	5.04	3.62	5.85
F	2.40	3.67	3.32	3.74

Kissling Resins.

(*e*) The method for the determination of the Kissling resins in an oil is somewhat similar to the method for the determination of the saponification number. The weighed sample of oil is boiled with alcoholic caustic potash; but, instead of titrating with standard acid, the alcohol caustic-potash solution is separated from the oil and made slightly acid with dilute hydrochloric acid. The resins are precipitated and are then separated from the acid solution by shaking with benzol. The benzol solution is evaporated in a weighed dish, and the resins remaining in the dish after the evaporation are weighed and the result reported as percentage by weight. The resins obtained by this test contain, besides the saponifiable resins, some resins which are soluble in alcohol solutions. In Table VII are given the results of this test on the original oils, the used oils, some of the original oils which have been subjected to the G. E. test and the original oils subjected to the Waters' test.

TABLE VII.

Percentage by Weight of Kissling Resins.

Name of oil.	Original oil.	Used oil.	Original oil after G. E. test.	Original oil after Waters' test.
A	0.68	1.07	. . .	1.85
B	0.57	1.19	. . .	0.76
C	0.20	0.36	0.51	0.53
D	0.42	0.36	0.91	1.26
E	0.36	1.46	0.60	0.82
F	0.44	0.56	0.85	0.51

Light Colored Resins.

(*f*) Resins extracted from lubricating oils by 70 per cent. alcohol are of a light yellow color and have a higher specific gravity and a higher oxygen content than the oil from which they are extracted. For the purpose of the test it was desirable to know if a lubricating oil when used in service would gain in its content of these resins produced by the oxidation

of the oil. The procedure for their determination was to treat 50 grams of the oil in a flask with 100 cc. of 70 per cent. alcohol; fit the flask with a reflux condenser and bring to a boiling temperature; the whole was then allowed to cool and one-half of the alcohol solution was separated off, placed in a weighed porcelain dish, evaporated to dryness and the resins weighed. The result was expressed as percentage by weight of the original oil. In Table VIII are given the results of the test.

TABLE VIII.

Percentage of Resin Soluble in 70 per cent. Alcohol.

Name of oil.	Original oil.	Used oil.
A	0.73	1.36
B	0.44	1.40
C	0.30	0.73
D	0.72	0.80
E	0.20	1.28
F	0.40	0.84

Resins Removed by Animal Charcoal.

(*g*) The method for the determination of heavy resins in lubricating oils was developed at this Station. At first trouble was experienced in obtaining the heavy resins due to their solubility in gasoline, but this difficulty was overcome by the use of ice-cold gasoline in washing the animal charcoal and resins, and by cooling the mixture before filtration. Since the refractive indices of the highly oxidized petroleum resins are much higher than the refractive indices of the oil, it was found possible by the use of the refractometer to vary conditions so that a separation of oil and heavy resins was obtained. The procedure of the test, as finally developed, is as follows: To 5 grams of the oil in an Erlenmeyer flask is added 25 cc. of 88-degree gasoline and 5 grams of dry animal charcoal; the flask is then tightly corked and allowed to stand over night in a cool, dark place. After this period the flask and its contents are cooled for one hour in an ice-salt cooling

mixture, filtered rapidly by suction on a perforated Gooch crucible with a dry asbestos pad and washed with 75 cc. of ice-cold 88-degree gasoline. The contents of the Gooch crucible are then transferred to a flask, 50 cc. of chloroform are added and the flask is placed on the steam bath until the chloroform starts to boil; the chloroform extract is then separated from the animal charcoal by suction filtration in a Gooch crucible prepared as above. The animal charcoal in the flask and in the Gooch crucible is washed several times with warm chloroform. The combined chloroform extract is then evaporated in a weighed dish and the resins weighed and reported as percentage by weight of the oil. To avoid the possibility of the resins becoming contaminated with the oil, also for the purpose of identification, the refractive index of the resins is taken at 20 centigrade. The index of refraction of the oil, free from heavy resins, is also obtained by evaporating off the gasoline from the oil in the first filtration and determining its index of refraction at 20 centigrade. In Table IX are given the percentage by weight of highly oxidized resins present in the original and in the used oils and the index of refraction of the resins found in the original and in the used oils.

TABLE IX.

Name of oil.	Per cent. of resins in original oil.	Index of refraction at 20 C. of resins in original oil.	Per cent. of resins in used oils.	Index of refraction at 20 C. of resins in used oil.
A	3.14	1.5550	4.42	1.5762
B	2.43	1.5573	3.75	1.5710
C	2.71	1.5560	4.06	1.5510
D	1.88	1.5570	2.81	1.5750
E	2.33	1.5488	3.60	1.5490
F	2.95	1.5658	3.73	1.5650

In Table X are given the indices of refraction of the used and of the unused oils, and their indices after the heavy resins have been removed.

TABLE X.

Index of Refraction at 20 C.

Name of oil.	Original oil.	Original oil less resins.	Used oil.	Used oil less resins.
A	1.5136	1.5066	1.5150	1.5100
B	1.5082	1.5040	1.5096	1.5046
C	1.4906	1.4877	1.4889	1.4864
D	1.5123	1.5097	1.5081	1.5045
E	1.4786	1.4778	1.4794	1.4778
F	1.4896	1.4864	1.4906	1.4866

Unsaturated Compounds.

(*h*) The procedure for this test is to measure out 10 cc. of the oil to be tested in a small 25 cc. graduated cylinder, add 5 cc. of kerosene, free from unsaturated compounds, mix, cool in ice water, and add from a graduated burette 10 cc. of concentrated C. P. sulphuric acid of 1.84 specific gravity; again cool in ice water. After standing for about 10 minutes in the ice water the cylinder is removed, tightly corked, and shaken violently for a few seconds; again place in ice water until thoroughly cooled, remove and shake violently for one minute. After standing a few minutes, the cork is removed, the cylinder is placed in a special laboratory centrifuge and revolved until the oil becomes clear and no longer increases in volume. It is then allowed to stand at room temperature until the total contents register 25 cc. The volume of oil absorbed by the sulphuric acid is then read off and is reported as percentage of unsaturated compounds. In Table XI are given the results of this test on used and unused oils.

TABLE XI.

Percentage of Unsaturated Compounds.

Name of oil.	Original oil.	Used oil.
A	12.0	18.0
B	14.0	20.0
C	12.0	15.0
D	12.0	14.0
E	12.0	16.0
F	12.0	15.0

Emulsification Data.

(*i*) It was desired to determine if used oils showed increase to any marked extent in the property of emulsifying with water. For this purpase equal parts of oil and distilled water heated to 130 F. in a 100 cc. graduated cylinder were stirred by the small copper paddle of the Station's emulsion machine for five minutes at the rate of 1,500 revolutions per minute. In Table XII are given the results of the test.

TABLE XII.

Emulsification at the End of 24 Hours.

Name of oil.	Original oil.	Used oil.
A	Trace.	Trace.
B	Heavy	Heavy.
C	None.	Trace.
D	None.	Noue.
E	None.	None.
F	None.	Trace.

Color.

(*j*) The color of the used and unused oils by reflected light was noted. Unfiltered oils or oils high in resins are brown to greenish black in reflected light. It was thought that it would be of interest to ascertain if this brown to greenish black color became apparent as the resin content of the oil increased in use. In Table XIII are given the results obtained.

TABLE XIII.

Color of Oil by Reflected Light.

Name of oil.	Original oil.	Used oil.
A	Dark with green tinge.	Greenish black.
B	Dark with green tinge.	Greenish black.
C	Green with dark tinge.	Dark Green.
D	Light yellowish red.	Greenish red.
E	Light reddish yellow.	Greenish black.
F	Red, slight greenish tinge.	Greenish black.

Physical Constants.

(*k*) In Table XIV are given the physical constants of the used and unused oils. Specific gravity was obtained by using the pyknometer. The Pensky-Marten closed cup and the Cleveland open cup were used for determining the flash and fire points. Viscosities were determined by the Saybolt standard viscosimeter.

TABLE XIV.

Oil.	Viscosity at—		Flash point in degrees F.		Fire point in degrees F.		Specific gravity at $\frac{60°}{60°}$	Baumé gravity.
	100 F.	130 F.	Open cup.	Closed cup.	Open cup.	Closed cup.		
Used A.....	690	254	382	345	424	486	0.939	19.10
Unused A.	512	204	380	344	430	360	0.930	20.50
Used B	312	138	362	.....	400	.....	0.927	21.03
Unused B..	310	138	366	340	416	370	0.924	21.50
Used C.....	316	149	.....	378	.....	424	0.885	28.19
Unused C..	290	144	.....	380	.....	422	0.878	29.45
Used D.....	315	140	370	286	410	325	0.920	22.17
Unused D..	348	148	362	334	394	372	0.931	20.38
Used E.....	160	90	408	376	448	416	0.872	30.55
Unused E..	139	65	398	390	451	436	0.866	31.66
Used F.....	187	102	404	364	440	410	0.888	27.66
Unused F..	178	100	410	403	448	424	0.882	28.73

Discussion.

Table III shows that a force-feed lubricating oil before and after use changes very little in its sulphur content.

Table IV shows that organic acidity of force-feed oils increases with use due to oxidation of the oil. The high acidity of used E is believed to be partly due to the addition of 200 gallons of old oil which had been boiled out at 325 F. (see Table II.) This high temperature used in boiling out the oil would, undoubtedly, assist in the oxidation of the oil and thereby produce acid resins. It is not believed that lubricating oil in contact with air at temperatures as low as 150 F., will oxidize or acidify to any extent, unless it is subjected to a severe strain as in lubricating machinery. This belief is based on the following experiments: Seven samples of unused

lubricating oil, three of which were oils used in this test, were heated at a constant temperature of 150 F. for 100 hours with air bubbling through each oil sample for this period. At the end of the test these samples were compared with samples before treatment. It was found that the organic acidity had not increased by this treatment in sufficient amount to be detected, and the treated oil samples had not in the least darkened. It is very evident from Table IV and from the tables following it that the strain on the oil film in the lubrication of machinery is a very severe one.

Asphaltenes in lubricating oils are formed by condensation or polymerization of the resins in the oil. The resins form the first stage in the conversion of unsaturated petroleum hydrocarbons into asphaltenes. By oxidation of the unsaturated hydrocarbons in lubricating oil resins are formed, and by the action of heat or pressure these resins are condensed or polymerized to asphaltenes. Indirectly asphaltenes are the result of the oxidation of the lubricating oil. As shown in Table V, practically the same amounts of asphaltenes are found in oils oxidized by the G. E. and the Waters' tests as are found in used oils.

The saponification numbers of the oils oxidized by the G. E. and Waters' tests, Table VI, point clearly to the fact that the increased saponification numbers of the used oils were due to their oxidation in service. A high saponification number for a used oil indicates that the used oil has greatly deteriorated.

The increase of Kissling resins in the used oils as shown in Table VII is also unquestionably due to oxidation.

As shown in Table VIII, the resins soluble in 70 per cent. alcohol, produced by oxidation of the oil, increase with the use of the oil.

The resins removed by animal charcoal are the highly-oxidized resins, with high specific gravity and high index of refraction. They vary from a hard, solid resin to a thick, gummy, sticky mass. They are generally of a deep brown to a black color, and are chiefly responsible for the

darkening of the color of the oil in use. After these resins are removed from the oil the oil is of a pale yellow color, and is known to the trade as "pale filtered." These resins are very closely related to asphaltenes, and are in all probability an intermediate step between the less oxidized resins and asphaltene. From experiments made at this Station it is believed that between these resins and the unsaturated compounds in a lubricating oil there are a great many different classes of resins, with oxygen content and index of refraction gradually decreasing from that of the resin removed by animal charcoal. The percentages of resins removed by animal charcoal and the indices of refraction of these resins are shown in Table IX. Table X shows by aid of the refractometer the effect on the oil of removing these highly oxidized resins. It is especially to be noted that the index of refraction of the used oil after the heavy resins have been removed is lower than the index of refraction of the original oil. When the heavy resins were removed from the used oil the color of the resulting oil was lighter than that of the original oil, and was almost identical with the color of the unused oil free from heavy resins.

The unsaturated compounds removed by sulphuric acid, as shown in Table XI, include also the resins in the oil. It is to be noted that the used oils contain more unsaturated compounds than the original oil; but if the used oils were subjected to filtration through animal charcoal or fuller's earth, the usaturated compounds in the filtered oil would closely approximate the amount of unsaturated compounds in the original oil. Since the action of sulphuric acid at 0 centigrade on unsaturated compounds is mainly one of oxidation and polymerization, and since the greater changes in lubricating oil during use are due to oxidation and polymerization, it is at once apparent that oils intended for force-feed lubrication should give the minimum percentage possible soluble in sulphuric acid at ice-water temperature.

The two tests developed at this Station, one for the determination of highly oxidized resins and the other for the

determination of unsaturated compounds, give very valuable information as to the quality of force-feed oils. The former shows the amount of resins that are liable to produce solid deposits under the least provocation, while the latter shows to what extent we may expect oxidation of the oil.

In Table XII is shown that the period of use of the oils in this test did not greatly increase their property of emulsifying with water. With a greater increase in the period of use, and consequent greater increase in the resin and asphaltene content of these oils, undoubtedly more marked emulsification with water would take place. In order to determine if filtration through animal charcoal would take out the emulsifying constituents in an emulsifying oil, 500 grams of oil "B," which is heavily emulsifying, were mixed with 10 grams of animal charcoal in a flask, shaken and then filtered. The filtered oil when tested in the Station's emulsifying machine with distilled water at 130 F., settled out completely without any trace of emulsion in 12 minutes.

In Table XIII are noted the color changes which the oil underwent during use. It is of interest to note that used oil "D," which contained the smallest amount of resins removed by animal charcoal, was of a greenish red color by reflected light, instead of a dark green or greenish black color.

CONCLUSIONS.

So far as can be judged by the use of chemical means, the changes that the above force-feed lubricating oils underwent in use were due to the presence of air in contact with the oil film, causing oxidation and polymerization, which were apparent through the formation of resins and asphaltenes. Since these resins and asphaltenes are removed from the lubricating oil by filtering through animal charcoal or fuller's earth, it is evident that the original usefulness and value of lubricating oil which has been in service for a considerable time can be regained by filtering through animal charcoal or fuller's earth.

NOTES.

TESTS OF COOLERS FOR LUBRICATING OIL.

In an article contained in the "Rivista Marittima," Mr. M. Boella, of the Italian Corps of Naval Architects, has recently dealt with experiments carried out with several types of coolers for lubricating oil. He states that forced lubrication, which was introduced with high-speed reciprocating engines of the enclosed type, has since been applied extensively to other types of engines, particularly steam turbines and internal combustion engines. Forced lubrication is now in general use also for auxiliary machinery on board ship, and this has led to special attention being given to oil-cooling apparatus.

The efficiency of such apparatus has a great bearing upon the satisfactory working of this method of lubrication. The cooling of oil is surrounded by a few practical difficulties, due to the physical properties of the oil itself. In the case of oil coolers, as in that of all apparatus based upon heat transmission, the fluid to be cooled has to circulate, and

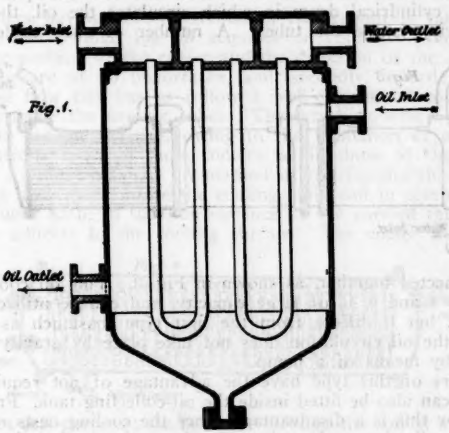


Fig. 1.

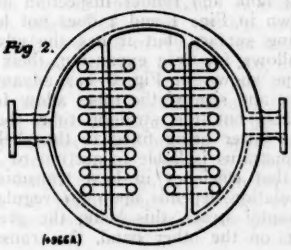
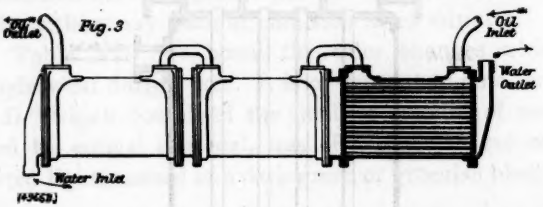


Fig 2.

to renovate itself, so to speak, by contact with a metallic wall, which is maintained at a low temperature by a flow of water or air. The viscosity of oil increases as the temperature falls, and there arises a tendency to the formation on the metallic wall of a more dense layer of oil, which adheres to it, and is not renewed, thus reducing the efficiency of the apparatus. The cooler must not merely have a sufficient surface, but special attention must be paid to the form of the surface and its arrangement, and to every detail which may contribute to the hastening of heat transmission. Moreover, in the case of oil coolers to be used for naval purposes, the space taken up, weight, continuity of action, and the facilities for inspection and cleaning, are also important conditions which have to be taken into account.

Figs. 1 and 2 show a vertical section and a plan of an oil cooler, the cylindrical receiver of which is generally inserted in the circuit traversed by the lubricating oil. It has a double cover, with partitions which divide the space between the inner and outer wall into three chambers. The cooling surface is formed by a number of tubes bent U-shape and fitted in the inner wall of the cover. Fig. 1 shows how the cooling water and the oil circulate through the apparatus; the latter can be emptied at any time by a valve at the bottom of the receiver.

Another type of oil cooler, which may be considered as an intermediary between those having a natural circulation and those in which circulation is a forced one, is illustrated in Fig. 3. This consists of a nest of small tubes welded at their ends into two tube-plates and contained in a cylindrical drum in which circulates the oil, the water for cooling flowing inside the tubes. A number of such coolers are fre-



quently connected together, as shown in Fig. 3. This oil cooler, like the one in Figs. 1 and 2, is of large capacity, and can be utilized as a collecting tank, but it differs from the first type inasmuch as it is horizontal, and the oil circulation does not take place by gravity but has to be effected by means of a pump.

The coolers of this type have the advantage of not requiring a collector; they can also be fitted inside the oil-collecting tank. From another point of view this is a disadvantage, since the cooling nests of tubes encumber the collecting tank and render inspection and maintenance difficult. The type shown in Figs 1 and 2 does not lend itself to a large development of cooling surface, but it has the advantage that the U-shape of the tubes allows for free expansion, their tightness remaining unimpaired. The type shown in Fig. 3 has advantages of a different kind; the arrangement and size of the tubes allow for a great extension of the cooling surface, but the straight tubes being welded to the tube plates, and these latter being fixed to the shell, free expansion is impossible, and the apparatus is liable to get out of order.

Mr. Boella states that efficiency in heat transmission in the coolers having a natural circulation depends upon the regularity with which the oil descends in horizontal strata, this being the greater the slower the flow of the oil. But, on the other hand, the transmission of heat be-

tween two fluids through an intervening wall is generally more active the greater the velocity of the fluid in contact with the wall. These two conditions are contradictory, and the effect is to limit the efficiency in heat transmission. In other words, coolers of the type under consideration are either of low efficiency, or else they should have a large cooling surface, when they become heavy and cumbrous, and consequently high in cost. For example, an oil cooler of the type shown in Figs. 1 and 2, forming part of an 8,000-h.p. marine turbine plant and consisting of a receiver 80 cm. (31.5 inches) in diameter and 1 m. (39.4 inches) high, acting at the same time as an oil collector, weighs complete, empty, 350 kg. (772 pounds) and has only 2.60 sq. m. (28 square feet) of cooling surface. The lubrication pumps have a discharge of a little less than 2 litres (0.44 gallon) per second. The efficiency of the cooler is therefore limited and the temperature of the oil remains high; the bearings of the turbine are, moreover, water-cooled. An horizontal cooler of the type shown in Fig. 3, used for cooling the lubricating oil of a marine 850-h.p. internal-combustion engine, and consisting of three equal lengths connected in series, 0.350 m. (13.8 inches) in diameter and 2.800 m. (110.3 inches) in length, weighs, complete, empty, 560 kg. (1,234 pounds) and has a total cooling surface of 45 sq. m. (484 square feet). This large cooling surface gives the apparatus a high efficiency; this, however, is obtained at the expense of weight and satisfactory working, since the very great number of tubes it contains (as many as 1,700) is a cause of frequent failures through leakage at the joints of the tubes with the tube plates.

In the coolers having forced circulation no account is taken of the increase in density which the oil acquires on cooling. Hence, in the first place, the position of the cooler and the direction of the oil flow inside the cooler are of no importance, and, secondly, in order better to govern the oil flow this has to follow a well-determined course, inside rather than outside the cooling tubes. The tendency which the cooling surface has to become inactive, owing to the formation of an inert oil stratum, manifests itself in these coolers, as in those of the preceding type, but for a different cause. A method of overcoming this is to give the oil flow a high speed inside the cooling coils and in compelling it to follow a tortuous path, so that the violence of the current thins the film of oil which adheres to the cooling surface. The cooler illustrated in

Fig. 4.

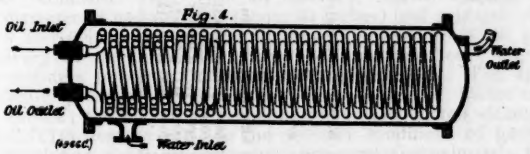
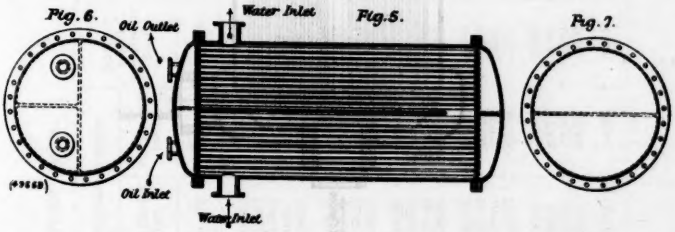


Fig. 4 answers this purpose; the oil flows through the coil at a speed of 4 m. to 5 m. (13 feet to 16 feet) per second. A cooler of this type for an internal-combustion engine of about 800-h.p. occupies a space the length of which is slightly under 3 m. (9 feet 10 inches) for a diameter of about 40 cm. (15.7 inches), and can transmit 200, and even 250, calories per hour per square meter of cooling surface and per degree Centigrade of difference in temperature between the oil and the cooling water (73 and 92 British thermal units per square foot per degree Fahrenheit). Such a result is a satisfactory one, but the space taken up by the apparatus is not negligible. Further, owing to the high speed of the oil flow and the great length of its path, a high resistance is opposed to the flow, necessitating the maintenance of a high pressure on the pump delivery. This has to be met by increasing the power of the lubricating pump.

Types of oil coolers.	Weight.	Diameter.	Height or length.	Space taken up.	Cooling surface.	Weight.	Space taken up.	Coefficient of transmission. Calories per sq. m. and Cent. degree and B.t.u. per sq. ft. per degree Fahr. of difference in temperature.
	kg. lb.	m. in.	m. in.	cub. m. cub. ft.	sq. m. sq. ft.	Per sq. m. kg. Per sq. ft. lb.	Per sq. m. cub. m. Per sq. ft. c. ft.	
Oil cooler, Figs. 1 and 2	350= 772	0.800=31.5	1.00= 39.4	0.502=17.7	2.60= 28	135 =27.6	0.190 =0.63	90 to 160= 33 to 59
Oil cooler, Fig. 3	560=1,234	0.350=13.8	2.80=110.3	0.270= 9.5	45 =484	12.5= 2.56	0.006 =0.02	.....
Oil cooler, Fig. 4	450= 992	0.400=15.7	3.00=118.1	0.380=13.4	20 =215	22.5= 4.61	0.019 =0.063	180 to 200= 66 to 73
Oil cooler, Figs. 5 to 7	700=1,543	0.600=23.6	1.70= 66.9	0.482=17	19 =204.52	36.8= 7.54	0.025 =0.084	70 to 95= 25 to 35
Oil cooler, Fig. 8 (two elements)	120= 265	0.320=12.6	0.40= 15.7	0.065= 2.3	5.76= 62	20.8= 4.26	0.011300=0.038	370 to 490=136 to 180

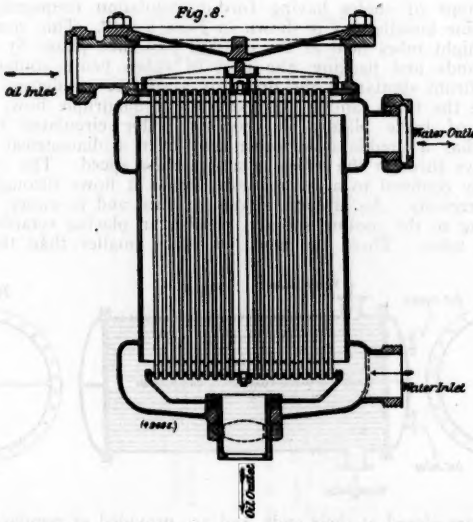
Another type of cooler having forced circulation frequently used in marine turbine installations is shown in Figs. 5 to 7. This consists of a nest of straight tubes held at the ends in two tube plates by means of screwed glands and packing, the nest of tubes being contained in a cylindrical drum similar to the drum of a surface condenser. The oil flows inside the tubes, and it has generally a quadruple flow, owing to the action of baffle plates; the cooling water circulates round the tubes, and has a double flow under and over a diametrical partition. The oil flows through the tubes at a moderate speed. The current of oil is mainly confined to a central core, hence it flows through without cooling appreciably. An attempt to remedy this, and to sweep along the film adhering to the cooling surface, consists in placing retarders inside the cooling tubes. These are made of tubes smaller than the cooling



tubes; they are closed at their ends, and are provided at regular intervals with transversal pins arranged so as to maintain them in the center of the cooling tubes. The device, although theoretically correct, does not give very good results in practice; it does not yield, according to the author of the article referred to, an efficacious lamination of the oil, and an imperfect centering of the retarders gives rise to the formation of paths of least resistance, to which the oil resorts in greater amount. This type of cooler has a most limited efficiency, much below that of the coil cooler, and also below that of coolers with free circulation for an equal cooling surface. A cooler of the type shown in Figs. 5 to 7—styled "condenser type"—for an 8,000-h.p. marine turbine, has an outside diameter of 55 to 60 cm. (about 23 inches) and a length of 1.7 m. (about 5 feet 6 inches). Its weight, empty, is 700 kg. (1,543 pounds). The delivery of the lubricating pump is 9 tons per hour; the cooling surface amounts to 19 sq. m. (over 204 square feet). The apparatus with difficulty produces a reduction of 6 to 8 degrees C. in the temperature of the oil; the maximum temperature remains in the vicinity of 48 degrees C. (118 degrees F.). In the ordinary conditions of temperature of the circulating water these results correspond approximately to a coefficient of transmission of 70 calories per hour per square meter of cooling surface and per degree Centigrade difference between the temperature of the oil and that of the water (25 British thermal units per square foot per degree Fahrenheit), a value which is indisputably low. From the point of view of space taken up and weight this type is not superior to the preceding ones, and since it is built throughout of brass its cost is higher.

An improved type of oil cooler is illustrated in Fig. 8; in this, adds Mr. Boella, the oil flow is laminated to a high degree by passing through a nest of flattened-down tubes; for this reason it is called a "lamellar cooler." The flat tubes are vertical, they are held in two tube plates, and are contained in a cylindrical drum in which circulates the cooling water. The tubes are short, thus facilitating manufacture. The upper tube plate is rigid, whilst the lower one is connected to a movable cover,

Fig. 8.



at the bottom of which is fitted the oil-outlet pipe, the arrangement ensuring the free expansion of the tubes and allowing the removal of the whole nest of tubes and its ready replacement by another set. The efficiency of this cooler is much higher than that of the preceding ones; it is obtained with a less cooling surface, and hence for a less weight and less bulk. A small apparatus of this type, weighing 37 kg. (81 pounds), and with a cooling surface of hardly 1.50 sq. m. (about 16 square feet), reduces by about 20 degrees C. the temperature of a current of oil of 2,000 kg. (about 2 tons) per hour, maintaining it at a maximum temperature of 60 degrees C. (140 degrees F.). The flattened tubes also lend themselves well to oil coolers in which the cooling medium is an air current. In these the oil cooler has the form of a circular arc, inside which revolves the ventilator fan.

The annexed table gives a comparison of the various oil coolers tested by Mr. Boella.

The experiments carried out were of a necessity limited in their scope and their results require confirmation; they, however, point to the fact that oil coolers are susceptible of important improvements. The influence which such appliances have upon the satisfactory working of lubricating plants, of which they form an essential part, amply justifies all efforts made to increase their efficiency.—"Engineering."

STANDARD CONDENSER-TUBE DATA.

By C. F. Braun.

The computation of surface condenser or other heat-transfer apparatus problems involving the length and number of tubes for a required heating or cooling surface, the weights per unit of surface or per tube, the velocity of flow of the water through the tubes, the relation of the internal surface to the external surface, etc., frequently becomes quite tedious.

The following table covering those sizes of tubes used in such apparatus should be very useful to engineers concerned with the design or selection of heat exchangers.

STANDARD CONDENSER TUBE DATA.

Outside diameter tube.	No. gage.	Weight per linear foot.	Weight per 100 sq. ft. outside surface.	Thickness.	Inside diameter.	Inside. Circumference.	Surface outside.		Surface inside per linear foot.	Inside sectional area per tube.	Velocity. per 1 G. P. M.	G. P. M. per 1 ft. per sec.
							Per linear foot.	Per linear inch.				
Inches.	S. W. G.	Pounds.	Pounds.	Inches.	Inches.	Inches.	Sq. Ft.	Sq. Ft.	Sq. Ft.	Sq. In.	Feet per Sec.	Vel.
	16	.421	256.7	.065	.495	1.559	.16362	.01364	.1299	.193	1.663	.602
	17	.380	232.2	.058	.509	1.598	.16362	.01364	.1331	.204	1.579	.635
⅝	18	.326	199.5	.049	.527	1.660	.16362	.01364	.1382	.218	1.472	.680
	20	.238	145.5	.035	.555	1.745	.16362	.01364	.1455	.242	1.328	.754
	22	.193	118.2	.028	.569	1.790	.19635	.01637	.1491	.254	1.265	.790
	16	.514	262.0	.065	.620	1.950	.19635	.01637	.1627	.302	1.069	.938
	17	.464	236.1	.058	.634	1.990	.19635	.01637	.1660	.3155	1.019	.983
¾	18	.396	220.0	.049	.652	2.047	.19635	.01637	.1706	.334	.9617	1.037
	20	.289	147.0	.035	.680	2.138	.19635	.01637	.1780	.364	.883	1.133
	16	.700	268.0	.065	.870	2.737	.26180	.02179	.2279	.595	.540	1.850
	17	.632	241.7	.058	.884	2.775	.26180	.02179	.2315	.6135	.524	1.910
1	18	.540	206.0	.049	.902	2.832	.26180	.02179	.2360	.638	.503	2.000
	20	.389	149.0	.035	.930	2.920	.26180	.02179	.2433	.679	.474	2.120
	16	.890	272.0	.065	1.120	3.520	.32708	.02725	.2935	.985	.3262	3.060
	17	.800	244.6	.058	1.134	3.561	.32708	.02725	.2970	1.011	.3172	3.150
1¼	18	.680	208.0	.049	1.152	3.620	.32708	.02725	.3020	1.045	.3075	3.255
	20	.490	150.0	.035	1.180	3.710	.32708	.02725	.3092	1.096	.2930	3.415
	10	2.110	537.5	.134	1.232	3.880	.3925	.03275	.3237	1.195	.2688	3.720
	12	1.75	445.5	.109	1.282	4.025	.3925	.03275	.3382	1.292	.2488	4.020
1½	14	1.36	346.0	.083	1.334	4.189	.3925	.03275	.3500	1.400	.2298	4.365
	16	1.080	265.0	.065	1.370	4.320	.3925	.03275	.3600	1.473	.2180	4.585
2	10	2.890	553.0	.134	1.730	5.440	.5233	.04360	.4534	2.350	.1365	7.330

* Weights given are for 60-40 brass. For pure copper multiply by 1.05.

NOTES.

THE SCOTTISH SHALE OIL INDUSTRY.

No. I.

We have on several occasions in the past, and notably in our issues of August 29th and September 5th and 12th, 1913, referred to the Scottish shale oil industry. The present time would appear, however, not to be inopportune again to return to this subject, seeing the importance which the supply of mineral oil and its derivatives has recently assumed. We propose, therefore, to publish two articles dealing with the industry. The first of these, which is printed below, treats the question generally, while the second, which will appear in a subsequent issue, goes more particularly into methods of manufacture.

The Scottish oil shale industry is remarkable, as it is the only British source which is producing oil and the various materials associated with or derived from it on a large scale, whilst the source of supply and mode of production are essentially different from those of other countries. Highly bituminous shale, containing about 80 per cent. of mineral matter, occurs in many parts of the world, as well as in the coal measures of the Lothians in Scotland. The shale oil industry originated in France two decades before it was introduced into Scotland. It is still carried on there to a considerable extent, principally in the four departments: Saône-et-Loire, Allier, Puy-de-Dôme, and Basses Alpes. The principal deposits of shale are found in the first two Departments. During recent years the French shale oil industry has not been so flourishing as the Scottish. Similar deposits of bituminous shales also occur in Serbia, New South Wales, Canada, and South Africa, but the industries associated with these deposits are on a very limited scale. There are, of course, other deposits of shale in the United Kingdom, but they are not at present being worked commercially.

This Scotch industry obtains its raw material from certain shaley rock strata, or deposits occurring in what is technically described as "calciferous" sandstones, at the base of the coal-bearing rocks, between the carboniferous limestone and the Old Red Sandstone. The mineral is obtained chiefly in Midlothian, Linlithgowshire, and in smaller quantities in Lanark and Sutherlandshire, while limited quantities have been obtained from Stirling and Flintshire. The large area of shales in central Scotland, extending south-west from St. Andrews and Dunbar to Ardrossan, in a belt 30 to 40 miles wide, does not entirely consist of oil shales. The latter have so far been found near West Calder, Uphall, Queensferry, Loanhead, and Bruntisland, over a distance of about 24 miles. The oil shale beds occur generally in small basins, cropping up to the surface on all sides, when not cut off by faults. They have in nearly all cases a small flat area in the center.

The thickness of the shale-bearing strata proved is about 3,000 feet. The following principal seams have been recognized in descending order;

(1) *The Dam or Raeburn Shale,* with an average thickness of 2½ feet to 3 feet. It has only been worked to a limited extent.

(2) *The Mungle or Stewart Shale,* average thickness 2½ feet.

(3) *The Grey Shale,* worked at Addiewell.

(4) *The Fell Shale,* the richest of the series. It is worked at Tarbrax, Addiewell, Breich, and Seafield. It produces from 36 to 40 gallons of crude oil, and yields from 25 pounds to 35 pounds of ammonium sulphate per ton.

(5) *The Broxburn Shale,* average thickness 5½ feet. It occurs at Addiewell, Seafield, Oakbank, Broxburn, Hopetoun and Philipstown. It produces 28 to 33 gallons of crude oil, and yields 30 pounds to 40 pounds of ammonium sulphate for every ton.

(6) *The Dunnet Shale* varies from 6 feet to 16 feet in thickness. It is found in West Calder, Oakbank, Pumpherston, Duddington, Newiston and Bruntisland. It yields 15 to 30 gallons of crude oil.

(7) *The Barracks Shale* has an average thickness of about 10 feet.

(8) *The Pumpherston Shales*, comprising the Jubilee shale about 8 feet thick; the Maybrick shale about 6 feet thick; the Curly shale about 6½ feet; the Big Plain shale about 8½ feet deep, and the Wee Plain shale about 4½ feet in thickness. All these strata are worked at Pumpherston, as well as at the Roman Camp Works of the Broxburn Oil Company. The Curly shale yields 19 gallons of crude oil per ton, and from 60 pounds to 70 pounds of sulphate of ammonia. On the average the Pumpherston shales yield 16 to 22 gallons of crude oil per ton, as well as 50 pounds to 60 pounds of sulphate of ammonia. The rocks lying between the various Pumpherston shales yield a large quantity of sulphate of ammonia, and are sometimes distilled together with the shales. The shale oil seams vary in thickness and quality in different districts. Thus the Broxburn shale, which yields over 30 gallons of oil to the ton at Broxburn and Oakbank, only produces 19 gallons at Linlithgow. Generally the upper seams down to the Broxburn shales contain more oil than the lower, the latter being richer in ammonia, which is a special feature of the Pumpherston shales. The workings have now reached a depth of 200 fathoms.

These shales are mined by the pillar-and-stall system when the seams are over 4 feet in thickness, the pillars being left large enough to be taken out in a second working. Seams under 4 feet in depth are worked by the longwall system. The shale is blasted down by gunpowder. It is then passed over a lin. riddle, the smalls being left underground in some cases. Water in the shale deposits is limited in quantity, and is raised by electrically-worked pumps. Although the shale evolves fire-damp, the mines are worked with naked lights, a very perfect system of ventilation being maintained by fans. The dust of this shale, unlike that of coal, is said to retard, instead of to propagate, explosions. The cost of production varies between 4 shillings and 6 shillings per ton of shale.

The crude oil of other countries occurs as a natural product. It either rises to the surface of the ground and spouts under pressure from below, as soon as a bore-hole is sunk, or it is pumped to the surface from underground reservoirs occurring in the folds of the rocks. The Scotch oil supply, on the other hand, is obtained by subjecting the shales to a distilling process, which yields fluids in the shape both of fuel oil and lamp oil, and, in addition, solid products, including paraffin wax, for match making and candles, sulphate of ammonia used as a manure, and other refined products.

The output of the Scotch shale oil industry before the war was annually approximately as follows:

Burning oils	20,000,000 gallons
Naphtha	5,000,000 gallons
Lubricating and gas-making oils	24,000,000 gallons
Solid paraffin	30,000 tons
Sulphate of ammonia	60,000 tons

The shale itself is dark colored, grey or black, with laminated or horny fracture, and usually has a specific gravity of about 1.75. As it generally contains 70 to 80 per cent. of mineral matter, it does not "coke" when heated to redness in a closed vessel. Twenty cubic feet weigh slightly less than a ton. Oil shales differ from coal shales in so far that the former yield members of the paraffin and olefine series, and the latter those of the benzene group. One million tons of Scotch oil shale when distilled

yield approximately 93,500 tons of crude oil and 19,000 tons of crude ammonium sulphate. After redistillation the former will yield about:

29,000 tons of illuminating oil.
5,500 tons motor spirit.
15,500 tons oil suitable for gas manufacture.
13,500 tons lubricating oil.
8,500 tons paraffin wax.

The crude oil has a dark green color, a specific gravity of 0.860 to 0.890, and is very viscous, owing to the large quantity of solid paraffin hydrocarbons which it contains, with a setting point of about 32 degrees C. It contains about 70 to 80 per cent. of members of the paraffin and olefine series, as well as alkaloidal bases derived from the pyridine series, and also some cresols and phenols in relatively small amounts. The average yield of the shale per ton is 22 gallons of oil and 45 pounds of sulphate of ammonia. Some of the seams worked are rich in oil and poor in sulphate of ammonia, whilst in others the reverse is the case.

The Scotch oil shale industry has been in existence about fifty years, and has steadily grown in importance and output, as shown by the following figures:

	Tons.
1873 the output of shale was	524,000
1882 the output of shale was	1,030,000
1888 the output of shale was	2,076,000
1910 the output of shale was	3,130,000

During the last decade the average has approximated to 2,800,000 tons annually. This quantity at 22 gallons per ton would yield 61,600,000 gallons of oil. Estimating the output of shale when we declared war at the rate of 3,280,000 tons per annum, the British annual yield of oil would reach 72,160,000 gallons. The average price of the shale at the mines slightly exceeds 5 shillings per ton. Recent outputs and costs have been as follows:

	Tons.	£	Per ton.
1908	2,892,000	795,200	5s. 5. 9d.
1909	2,967,000	815,900	5s. 5d.
1910	3,130,000	860,800	5s. 6d.
1911	3,116,000	857,100	5s. 6d.
1912	3,184,000	765,700	4s. 9. 7d.
1913	3,280,000	822,400	5s. 0. 1d.

The works and mines afford employment to about 10,000 men, and approximately a million pounds are paid annually in wages. Prosperity or depression in the Scotch mineral oil industry depends on the supply and prices of oil and sulphates, both of which are obtained from other competing sources. The cost of transporting these competing supplies is not entirely counterbalanced by the initial advantage of oil not having to be extracted from a solid raw material. When the solid products can be sold at a profit the oil may merely be produced as a by-product, and may, if necessary, be sold at cost price, or even at a considerable loss. The enormous output of natural crude oil during recent years from oversea oilfields of vast capacity has rendered its production from shale unprofitable, but its production is unavoidable, as the more valuable solids cannot be recovered from the shale without also producing the oil. When low prices for sulphate of ammonia and other solids synchronize with low prices for oil, the Scotch oil industry passes through a crisis, of which there have been several—notably one during 1911, which was little short of disastrous. It was very generally expected that this Scottish

mineral oil industry would benefit from an anticipated large demand for oil as fuel for marine engines and from an Admiralty demand for that purpose; but, unfortunately, the output of natural oils became so great that this expectation did not materialize. The importance of this native supply of fuel oil cannot be over-estimated. In case of emergency it would be invaluable. The industry ought to receive all possible encouragement and support, even if a price above the market level of the time has to be paid. The solid products obtained from the Scotch shales may also be recovered from some of the natural oils, but not from all. Thus, Russian and Roumanian oils do not yield hard paraffin or scale. Refined oil and its by-products are obtained from natural oil in such quantities that competition and price cutting has been ruinous. The increased freights during the war period have enabled the Scottish oil industry to recover some of the losses of recent years. During the last decade successful efforts have been made in Scotland to reduce the cost of production, but competition has neutralized them; they may tell, however, now.

The sulphate of ammonia which is produced from the Scotch shale is, of course, a reliable manure of great value. In addition to the competitive sources of supply mentioned above there are others. Sulphate of ammonia is obtained in large quantities as a by-product from gasworks, whilst supplies are obtained from by-product coke ovens and producer-gas plants. The world's output approximates to one and a-half million tons, of which probably one-third is produced in Germany. Great Britain comes next with a much smaller output, and the United States is a bad third. The French and Belgian outputs before the war were quite small. All other nations aggregated produced less than 200,000 tons. The quantity used in this country during recent years ranges from 90,000 to 100,000 tons, and our exports have been as follows:

	Tons.	£	Per ton. £ s. d.
1910	283,600 ..	3,376,700 ..	11 17 5
1911	291,100 ..	3,819,700 ..	13 2 5
1912	285,000 ..	3,991,200 ..	14 0 1
1913	323,000 ..	4,390,500 ..	13 11 10
1914	313,300 ..	3,767,500 ..	12 0 6
1915	294,300 ..	3,971,600 ..	13 9 11

The Scotch mineral oil industry is in the hands of four companies:—Young's Paraffin Light and Mineral Oil Company, Limited, established 1866; the Broxburn Oil Company, Limited, established 1877; the Pumpherston Oil Company, Limited, established 1883; and the Oakbank Oil Company, Limited, established 1886.—"The Engineer."

FEED-PUMP VALVES.

By A. L. Haas.

There is a matter concerning the effective lift allowable to feed-pump valves which is a case where practice contradicts theory.

The engineer to whom $\frac{\pi d^2}{4}$ and similar symbols concerning the values of the circle are familiar mysteries has always been through the demonstration that the lift of a valve should be ¼ d to give equal opening. Now, while the truth of this is not called into question, the expediency of allowing this amount of lift to every valve, irrespective of conditions, is a matter for question. Every valve of whatever type causes a loss of efficiency due to a factor impossible to do without—*i. e.*, that of friction.

The change in direction given to the flow of fluid causes a loss of head, and in the case of a feed-pump an increased pressure to overcome. The impediment of the valve seat has the same effect. For the last reason alone at least 25 per cent., and more often 33 per cent., must be subtracted from theoretical lift for physical reasons.

The trouble with text-book formulae is that in no case can they be applied except in the light of practical experience. They are absolute knowledge when of the $\frac{\pi d^2}{4}$ type, and most formulae of a practical order involve an empirical constant or factor of ignorance based upon actual performance of known result.

The designer of feed pumps pays more heed, like all hydraulic engineers, to the question of velocities rather than simple displacement volume. For example, in centrifugal pumps the whole question is one of velocity, not of direct pressure. A centrifugal pump is the only type which will run with less horsepower when the delivery valve is closed and involves no danger from excessive pressure to pump or pipe line. The reason is obvious. Under these conditions the impeller revolves as a solid flywheel, and no work is being done other than rotation. Speed in a centrifugal pump is based upon 10 feet per second flow, and most simple centrifugals are most economical at this speed. In other words, the speed of the extreme edge of the impeller is 10 feet per second, and a greater or less speed usually results in impaired efficiency. To obtain the quantity passed per minute it is merely necessary to calculate cross-section of delivery, multiplying by 10 and by 60; the result is displacement of pump. Another view of a centrifugal pump is that it is a booster, and high-lift centrifugals have the boosting feature in stages.

The bad feature of many common, simple centrifugals is that they absorb more horsepower with a low head and a consequently greater volume than when a higher head and lessened flow are the conditions of operation. For maximum efficiency it is necessary that the designed head and speed be kept constant. With a displacement pump, on the other hand, power absorbed is directly proportional to lift, and to close delivery with pump running may lead to disastrous results. For this reason relief valves are necessary, and invariably fitted in the pipe line or adjacent to the pump.

To return from the velocity pump pure and simple to the more usual type, a reciprocating boiler feed-pump of the duplex or other type usually has the delivery valve area more or less in accordance with the plunger area. This means that the velocity of flow through the pump valves is more or less the same as that imparted by the plunger; it is in a reciprocating pump of any description a series of squirts and not a steady flow.

Another matter is that, to be on the safe side, the normal feed pump should be much in excess of the duty required to furnish sufficient water for boiler evaporation. In fact, with feed pumps on the main engine at least 75 per cent. of their stroke is not pumping at all; in some cases it is forcing air into the boilers if the gland be not unduly tight. It is not overstating facts to say that the majority of feed pumps fitted are capable of supplying up to six times the normal feed as a maximum quantity. Most duplex pumps are rated on a moderate speed, which may safely be very much increased. The catalogue duties are based upon piston speed; in other words, their makers list them not from a displacement but from a velocity point of view.

Feed pumps of the type indicated are provided in most instances with small, flat valves, several in number, on both suction and delivery sides. If these valves are given theoretic lift the area of opening to flow leads to shock conditions in feed pipes. In the case of main-engine pumps from easily realized conditions the shock stress in the pipes is greater still,

and few are the steamers so fitted which survive the passage of years without repaired feed pipes.

If the lift of the valves be reduced and the delivery area be lessened the plunger of pump works at an increased pressure, and the velocity of flow through valves is increased. Such restriction of flow tends to lessen shock on pipes. Wherever feed pipes vibrate unduly it is pretty safe to reduce lift on pump valves. Experiment can determine the safe lift, and experiment alone; it is as well, though to put a pressure gage on pump chamber to ensure that conditions resulting are not too severe.

Bad cases of pipe shock have been cured by adjusting the valve lift to the thickness of a post card, and this without unduly stressing pump.

The smaller lift equalizes the feed water, which then proceeds in regular increments, and not in sudden large gulps. Want of this simple expedient has frequently led to the substitution of a specially-designed large, slow and deliberate pump of an expensive type to replace existing gear fully equal to all demands had the conditions been properly analyzed. In practice it will not be denied that most check valve regulation on boilers is severe upon the pump. Half a turn to check valve is usual. This regulation of feed effects results at the expense of shock to feed pipes. This, at least, is the present contention. A steadying effect is thus obtained at the wrong end of the system, which by adjustment of lift on the pump valves might have been more properly obtained at the beginning and not the end of the flow. The noise and vibration only too often found, which is a source of worry and discomfort to the engineer on watch, can be effectively altered much better in a large number of cases by diminishing the lift of the pump valves. If, therefore, feed pipes vibrate unduly it is as well to overhaul the lift of the delivery valves. In a number of instances a complete cure has been effected by a reduction in lift.

The whole question of valve design, whether stop, check, or when unadjustable, as in pump valves at work, can do with revision. Flanges designed more to save metal than give rigidity and spanner clearance are too much in evidence. So familiar a mechanical accessory, having years of precedent to fix type, has led to neglect of overhaul from the designer's end. Few are the valves which one comes across that cannot be adversely criticised; starting with the hand-wheel and ending with the joint faces, there is ample scope for improvement. Extra work is put where it is least necessary often enough, as in a highly polished hand-wheel which has to be served with yarn before a satisfactory grip can be obtained. Flange faces are made smooth so that the joint is less secure; a wire feed in machining is preferable to flange and a distinctly rough hand-wheel. Bolts and nuts are cheap enough to substitute six or eight for the too frequently noticed four. Smaller bolts and more of them are needed with the yielding type of flange in gunmetal, which is a perpetual sore place, not to mention ⅜-inch or ½-inch valves with oval flanges and two bolts.

The designer should be more generous in spirit in the weight of metal allowed. Most of the troubles experienced are impossible effectively to remedy when the job has been made up and is in being, causing daily annoyance to the engineer. The question of weak flanges alone has been a gold mine to the maker of patent packings and jointings; while the amount of skin lost from knuckles in the course of a year would cover an ordinary steamer's deck, laid (as the statisticians put it) end to end, from insufficient flange area and slipping spanners. One-thirty-second turn for each adjustment of the spanner to the nut is slow and wearisome work, and only too rarely are snugs fitted to the bolts. In small valves the wheels are too tiny for a big fist. There is a limit microscopically in this direction, while in large numbers of cases a wheel spanner whose

use is only too common testifies to the insufficient size of the wheels in larger sized valves.

The criticisms do not apply universally, since there are makers who exclusively deal with valves. Some of these turn out a decent article, but the ideal valve is yet to seek. Vast numbers of such fittings for boilers and elsewhere are the poorest of goods marketed for sale only. To protect their customers many engine firms make their own valves, and, since design is not a specialty confined to a few makers only, it is hoped the remarks here tendered may serve to give a pause for reflection and lead to better things. If the spirit of rational criticism is once evoked and draughtsmen and others responsible start to look upon existing practice with a critical eye there is little doubt that remedy will be effected. Where danger to life and limb are involved a small extra cost should not be allowed to stand in the way of safety first, which is a policy we all agree to subscribe to in spirit. No precaution is too costly to obviate troubles where risk is assumed daily, as where steam pressure is carried and where the remedies are known and easily applied it leaves little excuse after the event, if and when it happens.—"International Marine Engineering."

MECHANICAL DIFFICULTIES IN THE EVOLUTION OF THE STEAM TURBINE.

Staggered by the mechanical troubles which arose in connection with the development of one of the early designs of dynamos, an American investor is recorded as declaring that he wished he had decided to support a rival type, not realizing that the annoying and expensive incidents were the badge of all the tribe and were merely the inevitable accompaniment of pioneering work of this character. A few years ago agents for steam turbine builders quite frequently attempted to push their own wares by proffering long lists of the mishaps which had overtaken other makes, and from which they asserted their own particular type was immune. The only ground for this assertion, when, if ever, it happened to be true, lay generally only in the novelty of their own wares. "Happy is the country that has no history," and if only a few turbines of a particular type had been built the opportunities for creating a history of misadventures was inevitably lacking. This kind of salesmanship suffered from the drawback that when the new pattern, in its turn, met with misadventures, retaliation was inevitable, and as a consequence many engineers were led to consider the danger of breakdown of any type of steam turbine so serious as to make their adoption impossible, and many large reciprocating sets were installed during these years where the conditions were really such as to make the turbine a much more desirable machine.

In the early days, for example, it was asserted that blade stripping was confined to reaction turbines, with their necessarily small clearances over blade tips. Agents for impulse machines drew attention to the large clearances over blade tips which could be adopted with this type, and claimed, accordingly, complete immunity from this particular evil. Experience, however, soon proved that this proposition was *ben trovato* rather than *vero,* as strips with the earlier impulse turbines were so serious that at the Skodawerke, for example, almost the whole of the profits of the turbine department during the earlier years were swallowed up in replacing blades in turbines which had stripped before being taken over by their purchasers. Much expensive experience had to be paid for before both the design and the material of the blades became satisfactory. This circumstance shows clearly that stripping may result from other causes than small clearances. In fact, in the so-called disc and

drum machines it has often been found that the blades of the velocity-compounded wheel have been the first to go. This has been the case, at any rate, in power-station practice, where the temperature and the speed of the steam are both high. The matter, no doubt, is mainly a question of design and of materials. The suitability of either is hardly capable of determination by laboratory experiments. It is now ancient history that 25 per cent. nickel-steel proved a failure wherever used, and the A.E.G. have had disastrous experiences with 32 per cent. nickel-steel even when used for stationary guide blades. The soft brass alloy which has been so successfully used for Parsons' blading has been known to cut away inside of a fortnight when adopted for a velocity-compounded wheel.

A prolific source of blade breakages has been the chance synchonization of the natural periodicity of the blade, considered as a vibrating elastic solid, with some periodic impulse either inside or outside the turbine. This was, we believe, first observed with the exhaust blading of the first large steam turbine built by Messrs. Brown, Boveri and Company. In reaction turbines the fixed blades are commonly replaced by solid stoppers, at the joint of the turbine casing. There is thus an interruption, partial or complete, in the supply of steam to a moving blade whenever it passes the joint, and should the periodical time of passing these points agree with the natural period of vibration of the blade, breakage is inevitable. The remedy, which was successfully applied by Messrs. Brown, Boveri and Co. to the case mentioned, was to alter the rate of vibration by soldering an additional lacing wire to the blades at mid height. In the case of impulse turbines the use of partial admission stages also introduces a certain periodicity into the forces acting on the blades, and many breakages have been caused in this way. The remedy is, as before, a simple one, viz: to get rid of this synchronism between the varying force and the vibrator by stiffening the latter. Occasionally, however, this leads to curious results. Certain cases of failure having occurred from causes of this kind, the blades were stiffened by thickening the roots. This proved highly satisfactory until the stiffened blades were used upon a new size and pattern of turbine. With the introduction of this, breakages again appeared, but the blades broke not at the root, but at a point nearly one inch higher up.

The earlier turbines worked with saturated or but slightly superheated steam, and the introduction of higher superheats occasioned a plentiful crop of troubles. In the case of reaction turbines these commonly took the form of drums getting loose on their end shafts, due to the unequal expansion by heat of the body of the drum and of the stub connection on to which it was shrunk. This difficulty has long disappeared, owing to suitable modifications in the design. With both impulse and reaction turbines troubles arose through the distortion of the casing, owing generally to some lack of symmetry in the design or inequality in the rate at which heat is convected away from opposite sides. A hot casing will, in fact, hog considerably if uncovered. The air currents set up cool the under side more than the top, since they are already hot when they get there. Unequal expansion of this kind was a prolific source of trouble in the early days of the impulse turbines. At the best it meant a bent or broken shaft; whilst in other cases the trouble was still more serious, as in the case of the cruiser *Mainz*, to which turbines were fitted by the A.E.G. In this case the shaft welded itself solid to the diaphragms, through which, of course, the clearance had to be very small.

The difficulty arising from small shaft clearances was overcome by adopting Mr. Gerald Stoney's maxim that when one surface is to run at a high speed near another surface and with but a small clearance between, one of the two surfaces must be a mere thin edge. Once stated, the common sense of this principle appeals to all, and it has been universally

adopted by turbine builders the world over. Its discovery was, however, by no means either obvious or easy. Its general adoption has abolished any serious trouble from small clearances, but minor ones still occasionally appear. Two instances are known in which dummy grooves have been washed away by the leakage steam, leaving the metal everywhere fully ⅛ inch clear of the opposing dummy strips, for which nickel had been used. No explanation of this extraordinary occurrence has so far been found. Again, although the adoption of Stoney's principle has stopped the breakage of shafts by distortion troubles, the latter are probably responsible for the vibrations still occasionally experienced with two 7,000 kw. A.E.G. turbines. As originally delivered this trouble was so serious that the turbines were unable to take more than one-third of their rated loads, and had to be entirely rebuilt. This enabled them to take their loads if carefully operated.

With the increase in the sizes and speeds of turbine wheels another vibration trouble made its appearance. In certain cases, fortunately somewhat rare, the wheel itself begins to vibrate under the influence of certain forces which synchronize with its own periodicity. The periphery takes the form of a standing wave, which remains stationary in space whilst the wheel itself rotates. The wave consists of a displacement of the rim axially from its normal position when at rest. There are usually four wave lengths comprised in the complete circumference of the wheel. At four points the crests of the standing wave touch one of the two diaphragms between which the wheel is running, and at four intermediate points the troughs or "negative" crests of the wave touch the other diaphragm. Four points of wear are thus found on each. The result, unless the turbine is at once shut down, is that the rim gets red hot, wears away, and lets the blades out, causing a more or less complete strip. The clearance between the two diaphragms is commonly made as small as practicable, so as to shorten the overall length of the turbine, thus increasing the stiffness of the rotor. For the same reason the wheels are generally made thin, with a consequent reduction in their stiffness.

At one station two machines, both recent productions of the A.E.G., developed this trouble. On opening up one of the turbines it was noted that the offending wheel continued to vibrate in unison with some part of some other machine in the station, so that it appears possible that the trouble may have been started by something quite external to the turbine itself. In any case, the phenomenon is highly erratic. Two machines, both rated at 20,000 kw., were built from the same drawings. The first to be delivered has run without the slightest trouble ever since its erection. The other, whilst under test in the builder's shops, developed in one of its wheels the standing wave as above described, and had to be, to some extent, reconstructed. Possibly the difference in the behavior of the two machines may be due to slight differences in the rigidity of the joint between the wheel and the shaft. Two forced fits can never be exactly the same, and in the case of the A.E.G. machines already referred to it was found that the vibration was completely cured by clamping to the wheel a weight of a few ounces.—"Engineering."

LUBRICATION OF DIESEL ENGINES.*

By G. B. Vickers.

For the lubrication of Diesel engines pure mineral or hydrocarbon oils are undoubtedly the best; they contain a much smaller percentage of acid than animal or vegetable oils. The best test of lubricating oil is on the air compressor. If the oil causes pitting on the compressor

*From a paper read before the Diesel Engine Users' Association.

valves and shows an acid scouring action on the valves in the air bottle heads, the engineer may expect the main cylinder liner wear to be excessive. Average liner wear is 0.010 inch to 0.012 inch per 1,000 hours' run, when the engine is heavily taxed. For pistons, an oil with a moderate viscosity of, say, 130 to 180 at 140 degrees F., gives good results, whilst an oil with a good viscosity of, say, 400 to 500 at 140 degrees F., is best for small end lubrication.

For enclosed-type engines thicker oil is required than with the open type, as the temperature inside the crankcase is much higher than on open-type engines. In experiments on enclosed engines made by Messrs. Hick, Hargreaves & Co., Limited, difficulty was experienced in using thin oil which has a viscosity of 110 at 140 degrees F. Thicker oil was used, having a viscosity of 140 at 140 degrees F., in place of the thin oil, and the fuel consumption, which previously had never been lower than 0.45 pounds per B.H.P.-hour, was reduced to 0.419 pound, effecting a saving of 15 shillings in a six-hour trial run.

On average engines the quantity of lubricating oil used is kept as low as possible by using new oil on the small ends only, and filtered oil on the other parts. This filtered oil has advantages, as after running and filtering, the bituminous matter in the oil is eliminated. Filtered oil should not be used too often on the pistons, as its viscosity gets too low for this work. The original method of lubricating the piston by a single stem leading to a belt encircling the liner, and from thence through small holes in the liner, is a poor method. The belt should be flushed out with paraffin every time that the pistons are withdrawn, or perhaps even once a week.

The usual method of piston lubrication is through four or six stems or quills leading through the water jacket to the liner, and connected by a common feed pipe, past a back-pressure valve, to the single lubricating pump. Many engineers now insist on having a separate feed to each quill—*i. e.*, they have a 6- or 8-feed lubricator fitted for each cylinder. This is undoubtedly the better plan. The pumps for piston lubrication are in the majority of cases driven from the camshaft, and so are placed much higher than the quill line. The result is that, after a short stoppage, the feed piping has been drained, and the piston does not receive any lubrication for a few minutes after starting. A better plan is to have the lubricators fixed well below the lubricating belt line and worked off the indicator gear or the vertical shaft, thus ensuring that the pipes are always charged. The check valve in the lubricator pipe should be well designed, and periodically examined to ensure that there is no bituminous matter holding the valve up and interfering with the supply.

The method of securing the quill is sometimes the cause of trouble. Some makers prepare a tapered hole in the liner, and the quill has a rounded nose and is screwed home tight through a tapped hole in the water jacket. When the parts get warm and expand these quills have been found to act as struts, and have caused piston seizures. An improved method is to have the quill screwed into the liner, a plain hole in the water jacket, and an external joint.

Some makers have not adopted quills in the liner, but have relied on the splash system. This system is really only suitable for small engines, and its chief drawbacks are that lubrication cannot be regulated, is most uncertain, and there is a tendency to wastage.

Phosphor-bronze bearings are usually adopted for the top end bearings. The wear on phosphor-bronze bearings is seldom more than 0.001 inch per annum, whilst white-metal bearings, according to the quality, may show more wear than this in one week. When a phosphor-bronze split bush is used, the average clearance is 0.003 inch vertically and 0.006 inch at each side. If the bush is solid, more clearance is required, usually 0.006 inch to 0.008 inch vertically and 0.008 inch at the sides. The gud-

geon pins should be fully casehardened, at the ends as well as on the bearing surface.

A good long skirt is advisable to act as a guide and to reduce the pressure per unit area due to thrust. The length of guiding surface on the piston should be 1.4 to 1.6 time its diameter for low-speed engines, but may be reduced to 1.2 time the diameter for high-speed engines. The scraper method for lubricating the top end bearings, depending on collecting the oil remaining on the liner walls after the rings have scraped the bulk off, seems a very haphazard method, but it is quite effective for pistons up to 24 inches in diameter.

In the banjo system the oil is led into a banjo on the side of the crank, and the centrifugal force causes the oil to flow into a small receiver and pass up the pipe leading to the top end. Without a knowledge of the effectiveness of this system, any engineer would be justified in expecting that at least a special quality of oil, of the "monkey" brand, which would climb up the pipe, would be necessary for the system.

The most general method of lubrication is to have one or two slots in the piston which pass over the leads from the oil-supply pipes and holes leading from the bottom of the slots through the piston to the center hole in the gudgeon pin. Several holes lead from this central hole in the pin to the bearing surface. It is preferable to have two slots in the piston, one midway between the crankshaft center line and the cross center line on the front side of the piston, and one directly opposite, so that whichever side of the liner the piston is thrust against, one of the slots is able to scrape the maximum quantity of oil from the liner. The slot is best when V-shape in section, and the top and bottom should be undercut at about 45 degrees, the scraping edges being left moderately sharp; a sluice should also be cut to connect the top of the slot to the circular scraper groove cut in the piston to take advantage of the oil scraped off the walls by this groove. The hole in the center of the pin should be at least 1¼ inch diameter for a 6-inch pin, as it acts as a reservoir for the oil. Two holes at least should be drilled from the top of the pin to the center, one close to each end of the bearing surface at 30 degrees to vertical center line. These holes should lead into longitudinal grooves on the top surface of the pin, with well-rounded edges to assist the oil to escape.

On medium and high-speed engines a complete service of forced lubrication solves lubrication difficulties. Of course, this means having the engine fully enclosed. The practice of fitting a lubricating pipe leading from the bottom to the top end is to be deprecated; it is much better to have the oil passage up the center of the connecting-rod. When a priming system of lubrication is not provided, a small ball valve fitted in a cage in the hole at the bottom end of the connecting rod acts as a retaining valve when the engine is shut down, and prevents the oil from draining away from the top end.

Forced lubrication has disadvantages. Unless manufacturers have taken special precautions, it is probable that the piston will receive too much lubrication. The oil is thrown from the bottom end bearings on to the liner walls, and when the piston is on its suction stroke, the slight pressure in the crank chamber tends to force the oil past the relaxed rings, the result being that the lubricating oil is burnt and very peaky indicator cards are obtained, showing a maximum pressure frequently 100 to 150 pounds above compression pressure. The high consumption of lubricating oil has retarded the progress of the enclosed type of engine. We have overcome the difficulties, first by guarding the bottom end bearings to avoid splash on to the liners; secondly, by preventing the oil from creeping from the top end bearing along the gudgeon-pin keyway on to the piston surface; thirdly, by providing scraper grooves on the piston with return ducts to the inside; fourthly, by dissipating the vapor in the

crank chamber, this vapor tending to get past the rings on the suction stroke.

The common method of withdrawing the oil vapor is to take the air compressor suction, or the main cylinder suction, from the chamber. This causes dirty valves, is wasteful, and has proved to be dangerous. We find that the vapor is most effectively withdrawn by a belt-driven extraction fan, and we lead the gases to a baffle box where they are condensed, thus recovering the oil which by other methods is burnt. The vapor is mainly caused by the oil being splashed on the inside of the piston crown, and if the crown has an oil guard fitted the vapor is considerably reduced.—"The Steamship."

THE PROBLEM OF AEROPLANE ENGINE DESIGN.*

By Charles E. Lucke, New York, N. Y., Member of the Society.

The problem of the aeroplane engine appeals strongly to every engineer because it is a problem of the lightest power plant. The lightest weight of engine proper per horse-power is to be secured first by obtaining maximum mean effective pressure at maximum speed: in other words, the product of the mean effective pressure and the speed must be a maximum. At the same time the weight of metal per cylinder, or per cubic inch of cylinder displacement per working stroke must be a minimum, and with both of these factors the engine must be reliable in operation. So far, this reliability factor has been weakest, though lightness has been secured in engines good for short periods of running.

Not only must the metal weight of engine per horsepower be a minimum, but in addition the fuel weight to be carried must also be a minimum because, as can readily be seen, the fuel weight necessary for flights of any length predominates over the engine weight. For example: taking a half pound of fuel and oil per hour per horsepower as a fair value, it is readily seen how quickly that will catch up on engine weight when the latter is 4 or 5 pounds per horsepower.

In undertaking an analysis of the aeroplane-engine problem from the records, the only conclusion that can be drawn is along the line of type. Data are almost entirely lacking. On the question of general engine types, attention might be called to a few points:

The air-cooled motor has entirely failed in comparison with the water-cooled motor, the reasons are perfectly sound and secure. The 2-cycle engine has given way to the 4-cycle type.

Fixed cylinders have prevailed over rotating cylinders. Odd cylinder arrangements of queer, freaky forms have all been relegated to the scrap heap in favor of a few modern arrangements. The standard cylinder arrangements of today, which are the survivors of what may be called the inventive period, or at least the first inventive period, are the six and eight cylinders in line and the eight, twelve and sixteen V's.

It really appears therefore that the one valuable result of all our experience has been the selection of a few typical arrangements which we are now compelled to study, as minutely as circumstances permit, for the purpose of standardizing and mechanically perfecting these particular types as standard machines which will run as reliably as our stationary engines and which can be manufactured as economically.

Taking up each of the factors of aeroplane-engine design that seem

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important, in as specific a way as seems proper, the first one I wish to consider is the value of efficiency and the relation of efficiency to minimum weight.

Plotting hours of running as abscissae against weight of engine, with fuel and oil, as ordinates, for the air-cooled and the water-cooled types of motor, respectively, so that the intercept on the vertical axis represents the weight of engine metal alone, and the ordinates away from the axis represent the weight of metal plus fuel and oil, one finds that the two curves cross at some period of running beyond which, therefore, the water-cooled heavier engine, because of its lower fuel consumption, becomes lighter in comparison.

The metal weight of the water-cooled motor is about one and one-half times that of the air-cooled motor, and the slope of the combined-weight line of the latter compared with that of the former is as two is to one, that is to say: the consumption of the air-cooled motor is approximately twice that of the water-cooled motor. These facts are responsible for the crossing of the lines.

Of the conditions for efficiency which bear upon this question of fuel weight, and which have led to the selection of the water-cooled motor as a type, the first is the compression. The higher the compression the higher the efficiency, and there is no limit until preignition occurs. Statements will be found in textbooks to the effect that there is a limit, but they are the results of mistakes in interpretation, and are erroneous. The amount of compression possible is limited, however, by the metal temperature and by the temperature of the mixture as admitted. Naturally, the warmer the mixture during suction, the sooner it reaches ignition temperature by compression. Therefore, suction heating is a limit. Again, the interior metal temperature, if it is high (as it is always), may cause trouble by contact with the mixture during compression, and some portion of the mixture may be brought to its ignition temperature by hot-wall contact long before the main mass is brought to this ignition temperature by compression alone. It requires only one such hot spot to wreck a well-laid plan.

The next factor in efficiency is the mixture quality, and in this there are the following controlling elements: first, mixture proportions. Any excess fuel means direct waste, but it also means carbonization and fouling. Excess air quickly makes the mixture practically non-burnable. Therefore, mixture proportions must be accurately controlled—more accurately than is possible with any existing carburetor. Carburetors are not yet satisfactory, and as soon as satisfactory carburetors are secured from the standpoint of proportionality of the mixture, we may expect to see a further reduction in fuel consumption and more reliable operation.

Dryness of mixture is a matter of coördinate importance with mixture proportions. When mixtures are wet, that is not completely vaporized, the air and fuel cannot be uniformly distributed to the various cylinders by the manifold system. One cylinder will get a different charge from another, as can be easily proved by pressure gages. There are rarely two cylinders alike as to maximum pressures on a multi-cylinder engine using wet mixtures. Drying of the mixture will cure that fault, and also cure the carbonization that comes from the vaporization of the liquid in the presence of the burning gas when it has been admitted to the cylinder in a liquid state.

The third factor of the mixture question is homogeneity. However accurately the mixture may be adjusted as to fuel and air ratio, however carefully the mixture may be distributed, cylinder to cylinder, the fact remains that, in order to produce economical results, the charge in any one cylinder must be uniform in every cubic inch of it. It is not suffi-

cient that the right amount of air be in the cylinder even if the fuel is vaporized when the latter is all in one corner.

Following mixture quality, the next factor in efficiency is rate of flame propagation with reference to piston speed. It can be shown that the explosion line of the indicator card following compression must be maintained vertical for maximum efficiency. Now, the rate of propagation is the one factor that tends to hold it vertical. If the propagation rate is high enough for a given piston speed, so that the explosion line is vertical, the efficiency will be high. But should the piston speed exceed a certain value, then the explosion line will begin to lean toward the expansion line, until by and by it becomes horizontal and merges into the expansion line, with a consequent large loss of work area and low efficiency or high fuel consumption. Therefore, there is for every given mixture a limiting piston speed that cannot be exceeded without destroying efficiency, and we are now approaching that speed in aeroplane engines.

The next related factors are mean effective pressure, and speed. These are the prime factors for the output of a cylinder.

If the mean effective pressure were constant, then horsepower with reference to speed would follow a straight line. The mean effective pressure is not constant as the speed varies, however. Therefore, plotting horsepower against speed gives a curve having the general form of concave downward and consisting of several separate portions, each worthy of study. There is usually a straight portion over a given speed range, during which the mean effective pressure is constant. For lower speeds the mean effective pressure is lower, and for higher speeds the mean effective pressure is again lower. From the point where, with increasing speed, the straight line becomes a concave-downward curve, the mean effective pressure is decreasing as speed increases, until at the point where the tangent to the curve becomes horizontal, the rate of increase of speed is exactly equal to the rate of decrease of mean effective pressure. At a little higher speed mean effective pressure decreases faster than speed increases, and finally the curve drops down toward zero power.

So much for the facts. An analytical engineer cannot be content with those facts, however, but finds it necessary if he is to apply a cure to go behind the facts to ascertain the reasons. The first step in doing that is to determine the volumetric efficiency of the engine by measuring the air and fuel, and comparing the total volume of mixture taken in, with the piston displacement. If the volumetric efficiency be plotted against the speed much light is thrown on the situation. In the first place the volumetric efficiency falls off in the region of very low speed, where the mean effective pressure is low; it is constant over the region of constant mean effective pressure, where the horsepower speed line is straight, and then at some high speed it again decreases. It is clear, therefore, that curvature of the horsepower-speed line is due to a corresponding variation of volumetric efficiency. It may be found, however, that at some high speed the horsepower-speed line falls before the volumetric efficiency. This calls attention to the fact that the falling off of mean effective pressure at high speeds may not be due primarily to volumetric efficiency but to other causes, and recognition of this starts a search for those causes.

The first of these causes is too slow a combustion or too high a piston speed. That is to be corrected by adding an additional ignition source or by moving the spark plug from a side wall to a center point. Igniting at more than one point or at a more central point will cure this defect, and again cause the dropping points of both horsepower-speed and volumetric efficiency-speed curves to lie on the same speed line.

Again, it will be found that a change in the valve-setting changes this

mean-effective-pressure curve at both ends, but every change in the valve setting also changes the mean effective pressure, and the volumetric efficiency is itself the direct measure of whether or not one has the best valve setting.

Now, it is curious that most people have played with cams and adjusted them back and forward by guesses, and have never bothered about the air meter, which is the only positive means of arriving at best cam forms and valve timing for sustained mean effective pressure at high speeds.

Many more analyses along the above lines could be given, but enough has been said to call attention to this most important means of studying the problem of maximum power at high speed, not only revealing what is the matter but pointing out clearly the direction in which to correct the fault

So much for efficiency and mean effective pressure, or efficiency and horsepower per cubic foot of cylinder. Those two factors bear directly on the fuel weight to be carried and the output per cubic foot of cylinder. What will be the weight of that cubic foot of cylinder? This has to be judged both by qualitative and quantitative analysis. It is impossible to give any quantitative analysis without long mathematical treatment, so I will undertake only the qualitative analysis.

The first point in qualitatively analyzing unit metal weight of the multi-cylinder engine is to recognize that the engine can be divided laterally by planes into sections of one cylinder each. The end sections are the same as each other, but are different from the intermediate sections. Therefore, to study qualitatively the relative weights of two typical constructions, the mind must be concentrated upon these sections, each one of which includes a cylinder, a piece of frame, a piece of shaft and the other parts that go with the section.

From this point of view, consider multiplication of cylinders in line vs. radially or circumferentially. It will appear that the weight of the cylinder, piston and connecting rod, is just the same no matter how the cylinders are arranged, but the frame weight and shaft weight are reduced by any multiplication. It is clear also that, other things being equal, the lighter arrangement is circumferential rather than longitudinal multiplication.

Now, going back to the history of the situation, we find every conceivable combination has been tried, but these have finally crystallized to not more than two kinds, giving the V-type engine and the engine with cylinders in line.

Considering the effect of cylinder diameter upon unit metal weight, it will appear that from the unit-weight standpoint the cylinder diameter should be as large as possible, because the wall thickness of a cylinder is always greater than necessary for the stress for other structural reasons. A 1/16-inch cylinder of steel will not be stressed over, say, 10,000 pounds per square inch. The cylinder could be made much thinner than this and still have a good working stress if there were not other structural objections to it. This being the case, the larger the cylinder for a cubic foot of displacement the less the unit metal weight in the wall, and the only limit to large diameter is good running.

Considering the stroke, as this is increased the metal in the cylinder piles up endwise, or axially, too fast with reference to volume, and therefore for minimum unit metal weight, the shorter the stroke the better. In proportion, we are using, normally, shorter strokes in aeronautical motors than in automobile engines for that reason.

Again, as affecting the metal weight, we have the connecting rod length. Clearly, the shorter the connecting rod the shorter the frame, and therefore the more metal saved. The only objection to the shorter

connecting rod is an excessive angularity, which introduces stresses requiring metal thickening in other places.

The number of cylinders should be as large as possible up to the point where the weight of the connecting parts has to be increased. A 2-cylinder engine has less than twice the weight per cubic foot of displacement than a single cylinder, for the reason that the number of end supports for the shaft, etc., is not increased. Similarly a 3-cylinder has less than three times, a 4-cylinder less than four times, and so on; and the weight per cubic foot of displacement gets less and less until a certain number of cylinders—somewhere about six—is reached, where the shaft diameter and the weight of the frame must be increased so as to retain the necessary stiffness, whereupon the saving in weight by multiplication is neutralized. This appears to be about the limit of saving by line multiplication.

The metal weight per cubic foot of cylinder displacement has to be taken up along the lines indicated, extending the study to the form vs. weight of each individual member. It will appear, as one examines the forms of these individual members, that one form is clearly susceptible of less weight than another—even with the same working stresses or with equal factors of safety.

The first of these studies should be undertaken with reference to cylinders. The first cylinders built were made of cast iron, with head, cylinder and jacket cast in one piece, and the valves being arranged in a side pocket—the ordinary T- or L-head construction. It is clear that the weight of the valve pocket is detrimental. The first step in any cylinder-weight reduction, then, is to take that pocket away, retaining the cast cylinder (on the assumption that we do not know how to make any other kind) and putting the valve in the head. This results in the valve-in-head construction, which is now practically universal, but which, strange to say, it took six or seven years to realize.

A similar instance of slow realization of facts exists with reference to the cast-iron jacket wall, which has no other function than to hold water. Cast iron for that purpose, especially in an aeroplane engine, is wasteful of material, so the next step is to get rid of the cast iron. When one stops to think how it is to be done, a structural difficulty becomes apparent, and therefore one must not too readily condemn the holding on to the cast-iron jacket. The difficulty is, of course, the necessity of providing openings for the intake or outlet from each valve, an igniter-plug hole and at least two pipe connections for the jacket, and in an aeronautical engine under heavy stress there is some driving-gear which requires fastenings. This naturally tends toward the use of a casting.

Suppose such a casting is used, with inlet and one exhaust valve each with a port leading out, and such valve seating in the head which turns down to form the cylinder; then the casting may be led around the top, forming the enclosure of the head jacket and joining the several outlets and coming down outside the cylinder. The cylinder-head jacket casting ends in the form of a skirt at about the level of the valve deck, and to this end a tube jacket can be added by any one of several possible fastenings. That is the next step; cast iron for the cylinders, head and head jacket in a one-piece casting, but with sheet metal for the jacket over the cylindrical barrel. It is a logical step, but it took several years to reach it just the same.

Proceeding along the same line of weight reduction, the next step is to cut away this cast iron joining the ends of the ports and forming the wall of the head jacket, and substitute sheet metal welded to the ports by the oxygen welding system. Wherever there are connections to be made for attachment of gears there must be some additional supports welded or brazed on. The cast-iron cylinder is still there, and with cast-iron ports.

There is a fundamental objection to a cast-iron cylinder for aeronautical work, and it is a perfectly valid one. Cast-iron cylinders do not have to be very thick to be amply strong, so far as the gas-pressure stresses are concerned, but the fact remains that so long as they are cast iron, no one knows whether they are good cast iron inside or not, and the use of cast iron cut down to ⅛ inch in thickness incurs taking some chances. Hence attention is turned toward steel.

Drawn steel or forged steel is a reliable material and a logical selection, so designers have sought means of using it; but when one stops to think how to use a drawn-steel tube for a cylinder, and get the necessary attachments on it, one soon recognizes that the matter is not so easy as it looks. That is the reason the adoption of the steel cylinder was so long delayed.

There are now several schemes developed for steel cylinders. The first of these is a steel cylinder of a drawn tube formed without a head, screwed into a separate head carrying the ports and the head jacket cast in one piece. This is rather a satisfactory way of attaching a head, but it involves more than one difficulty. When such a screwed head is set up against the shoulder it is not at all clear just where it is going to stop; and to secure the proper position one must either scrape the faces or shim them—neither of which is a nice job. A further objection is the considerable weight of the cast iron in a rather complicated casting, and also the inner wall of that cast iron is a stress wall, the stress of which must pass through the thread to the cylinder. There is no objection to using a casting if it is not stressed, but a casting under stress is not satisfactory and is to be retained only in the absence of something better.

Complete elimination of castings has been tried by using all steel and sheet metal welded together, but this did not prove satisfactory for a very interesting reason. A flat sheet-metal head on which the valves are seated will not remain flat, and a round valve seat will not stay round. Such sheet metal tends to warp out of shape, and with it the valves will not stay tight. However, the material does not break, which is something worthy of thought.

To eliminate the weld between the steel cylinder and head, another construction was developed. In this, a seamless drawn-steel shell with head just like a cartridge is used, and two holes are arranged in the head to seat the valves. It is evident that this is a structure which is sound against all kinds of stresses. It still has some of the difficulties of warping the seats, causing leakage of the valves; and when a valve leaks the amount of heat developed is tremendous. Once a valve starts to leak, it is only a question of a short time before it will be completely destroyed.

The particular construction of cylinder just described is rather difficult to attach to its jacket ports. It is interesting to note one case at least in which a satisfactory attachment has been worked out, and that is the Hispano-Swissa engine, now used on the European war front, and now also being built in this country. In this particular engine the entire outside of the cylinder is threaded, and the cylinders are screwed into an aluminum casting which is double-walled just like the cast-iron block casting of an automobile engine. The thread performs the double purpose of holding the cylinder in place and bringing its head up against the aluminum cast head which carries the ports, and also acting as a thermal bridge between the metal of the cylinder and the metal of the aluminum casting which carries the jacket water. Without the latter there would be poor thermal contact and overheating of the cylinder. While this construction is not entirely satisfactory, it is nevertheless very interesting and suggestive. It immediately calls attention to the fact that a water jacket may be made of an aluminum casting and the ports formed just as easily as in iron, the steel interior carrying the stress due to the interior gas pressures.

It is, however, quite feasible to get rid of the double aluminum wall down along the cylinder barrel into which this steel cylinder is placed and which carries the ports above, by leaving out its interior wall and retaining the outside, or even by stopping the wall just below the head as a skirt to take a short thin tube which may itself be of aluminum, ending at the bottom in a cast stuffing-box ring to act as a joint against the steel cylinder. That, so far as I know, represents the last word in this direction, the steel cylinder head being bolted up to the aluminum head-port casting at the valve-seat bases, and not just pressed up against it by a remote thread.

Finally, there is to be noted the one-piece steel-forging construction for cylinder, cylinder head, ports and ignition holes, surrounded by a sheet-metal welded jacket, a very satisfactory though expensive construction.

These heads are themselves a subject of considerable study. We have first a plain head in which the valve inside diameter is half the cylinder less the width of seat, and half the bridge between the valves. Both valves have stems pointing upward and parallel. The plain cylinder, then, which can be made of a plain seamless-drawn steel cartridge, and which is so desirable structurally, limits valve diameter, and this is a factor against it. Valve diameter is a strong influence in volumetric efficiency and weight of charge, controlling, as it does, flow-resistance conditions. Naturally, designers must get the volumetric efficiency as high as possible by keeping flow resistance as low as possible. Therefore, the tendency is to go towards larger valves than is possible with the previous arrangement.

One variation in form for this purpose is the flat bulged head where the valve diameter is larger than before by the amount of the bulge. The flat bulged head is a very desirable thing for larger volumetric efficiency and higher mean effective pressure, but offers some difficulty in manufacture when one is making a one-piece seamless drawn-steel job, but not a serious difficulty.

Another suggestion for getting the same result is to bulge this head upward in the form of two flats and put the valves on the two inclines. It is perfectly clear that a very large increase in diameter can be secured in this way. The valve stems in this case are not parallel but diverge at any angle and the limit is reached when the angle is 180 degrees, in which case they are horizontal.

The question of block arrangement of cylinders and their jackets vs. separate units deserves some attention. In some cases each cylinder with its jacket and head is entirely separate. In other cases the jackets are cast or welded in a block form, around more than one cylinder—sometimes two and sometimes four, and in some cases six. It is clear that the more cylinders included in the jacket block, the less will be the weight of the jacket, because the length of the tangent to two jacket circles is less than a half circumference. But there are objections to the block, and in some cases it may not pay to use it.

In a case in point, a cast-aluminum block jacket was set down over four steel cylinders which were bolted to the frame by their usual flanges and studs. These cylinders gave trouble on the outer flanges, the end studs breaking off or pulling out. The trouble was caused by the crankcase running hot, expanding; and the aluminum-block cylinder casting running cool, because it was water-jacketed, not expanding. The cylinders being bent inward tore the stud ends right out.

Another point: the steel cylinder is naturally flexible, and it belongs—in fact the entire motor belongs—to that class of structures which should properly be termed flexible, exactly similar to bridge structures.

These flexible motors weave just as the engine of a steamship weaves. To attempt to hold one against springing is to attempt what is practical-

ly impossible. The cylinders of aeroplane engines should all be perfectly free to go as they will, and not be held on the top in any way. All the block arrangements of cylinders of the sort just described are therefore objectionable.

Steel cylinders have a natural spring and give to them, and if let alone they will serve well, but attempting to secure them may result in serious distortions, or in highly localized excess stresses.

Proceeding in the same qualitatively analytical style, the complete paper takes up successively the problems of the piston, valves, valve gears, frame, etc., indicating how a gradual realization of conditions has led to modifications of design, and pointing out factors not even yet considered and making suggestions for meeting them. For instance, in connection with the valves the author makes a thermal study of the problem which, he says, has not been undertaken by anyone in the shops. He does the same for the piston, and the crankcase he analyzes from its consideration as a stress member.

The paper sums up with the pointed statement that the aeronautical engine is emerging from the stage of invention to the stage of design; as a light, high-tensioned steel structure, consisting of seamless tubing and forged or welded steel parts, possibly formed in drop-forge dies. Add to that steel stress structure certain members, such as the piston, exhaust valve and guide, designed primarily for heat-flow conditions and not for stresses. Add to that again certain closing members, such as the ports for the intake and exhaust, which can be very properly cast in aluminum; and the oil crankcase closure, which can be made of any desired materials.—"Journal Am. Soc. M. E."

THE ANOMALY IN SUPERHEAT CORRECTIONS.

The rapid strides by which the design of electrical machinery was brought to its present stage of perfection is generally, and no doubt rightly, attributed to the relative ease with which theory could be applied to practice, and much of the equally striking development of the steam turbine is to be credited to a similar cause. In both cases the main difficulties have been practical rather than theoretical in character, but at the same time there was a much larger gap between theoretical and applied thermo-dynamics than existed between the electricity of the laboratory and class-room and that of the power station. Unexpected anomalies were found, for example, in the behavior of steam discharged from nozzles, since in certain cases the weight passed was proved to exceed the maximum possible on the then accepted theory of the phenomena. This difficulty was only cleared up three or four years ago, by the discovery that the steam was in a supersaturated condition as it passed the throat of the nozzle.

A still outstanding anomaly is to be found in the fact, on which all turbine builders are agreed, that the gain due to the use of superheat is very substantially in excess of what would be anticipated from the theory usually accepted. Of course, neither of these anomalies have had any material effect in retarding the development of the steam turbine. Certain experimental coefficients have to be introduced in any case, in order to pass from theoretical to practical thermo-dynamics, and by suitably choosing these the discrepancies between theory and practice can be rendered innocuous. Such a condition of affairs cannot, however, be considered as wholly satisfactory, since most of us are not content to merely observe and record sequences of events, but have an instinctive desire to fit all natural phenomena into a theoretical frame-

work, one advantage of which is that it becomes no longer necessary to memorize individual results, but merely to learn the general theory from which all possible results should be capable of being deduced. Attempts have accordingly been made to find some theoretical basis for the observed anomalies in superheat corrections. Part of them are due to the use of steam tables based on the erroneous experiments of Knoblauch and Jakob, but only a small proportion of the discrepancy can thus be accounted for. The usual explanation put forward is that wet steam meets with a much higher frictional resistance in flowing through the blading than does dry steam. This view is, however, incompatible with experimental observations on fluid friction. Professor C. H. Lees, F.R.S., has shown* that the whole of the best experimental results on the head lost in the turbulent flow of fluids through pipes can be represented by a single formula, which applies equally whether the fluid be air or water, and that at high speeds the effect of differences of viscosity are relatively small. Still earlier, Osborne Reynolds† had arrived at a similar conclusion, which he expressed as follows: "When the motion of a fluid is such that the resistance is as the square of the velocity the magnitude of this resistance is sensibly quite independent of the character of the fluid in all respects save that of density." In other words, if the velocity of motion be sufficiently high for the flow of the more viscous fluid to be highly turbulent, the head lost in a given length of pipe will be much the same if one fluid be treacle and the other water. These considerations show the inadequacy of the explanation usually offered for the apparent abnormally high efficiency observed with superheated steams. Another solution is accordingly required.

A fundamental theorem in thermo-dynamics established by Carnot is that the efficiency of a reversible heat engine working between two stated temperatures depends solely on these two temperatures, and is quite independent of the character of the working agent used. A reversible engine is one having an efficiency ratio of 100 per cent., and it has commonly been assumed that the theorem still held for actual heat engines, the efficiency ratio of which may be anything between say 50 per cent. to 80 per cent.

In comparing the observed gains due to the use of superheat with those theoretically due, the theory adopted has been based on the assumption that the effective thermo-dynamic head was proportional to the adiabatic heat drop. This effective thermo-dynamic head is, however, equal to the adiabatic heat drop multiplied by a "reheat factor," due to the fact that all the energy expended by friction in the higher stages of the turbine is returned to the steam as heat, and this addition of heat increases the amount available for doing work in the later stages of the turbine. It is easy to show that this reheat factor‡ is not independent of the character of the working fluid. As indicating the practical importance of this fact consider the case of a turbine supplied with steam so highly heated that the superheat is not lost before discharge to the condenser takes place. Then if the ratio of the initial to the final pres-

* Proceedings of the Royal Society, vol. xci, 1914.
† "Scientific Papers," vol. ii, page 237.
‡ For gases it is given by the relation

$$R = \frac{1}{\eta} \cdot \frac{1 - \left(\frac{p_2}{p_1}\right)^{\eta\left(1 - \frac{1}{\gamma}\right)}}{1 - \left(\frac{p_2}{p_1}\right)^{1 - \frac{1}{\gamma}}}$$

where p_1 and p_2 denote the initial and final pressures respectively, γ is the specific heat ratio, and η is the hydraulic efficiency of the turbine. This formula is not applicable to wet steam, but an exact method for use in this case was described in "Engineering," vol. 79, page 38.

sures be 200 to 1, and the hydraulic efficiency of the turbine 70 per cent., the reheat factor will be 1.1643, or, in other words, the effective thermodynamic head will be about 16½ per cent. greater than the adiabatic heat drop. If, however, we take another turbine, supplied with saturated steam at 200 pounds absolute and exhausting at 1 pound absolute, the reheat factor corresponding to the same hydraulic efficiency will be only 1.0559, or say, but 5½ per cent. more than the adiabatic heat drop.

The physical explanation of the reheat factor being higher with certain gases than with others may be of interest. The energy lost in friction in the higher stages of a turbine is returned to the working fluid as heat and causes the pressure of the fluid to be higher at the later stages than it would be at the same specific volume if there were no frictional losses. When heat is added to gas at constant volume, the corresponding energy goes in part in increasing the average velocity with which the molecules move, and in part to increasing the rate at which these molecules spin, and it may be also in part to intensifying intra-molecular vibrations. The pressure developed by a gas depends solely on the velocity with which the molecules move, and not at all on the rate at which they spin or at which their component parts vibrate. Hence only that fraction of the heat energy supplied which goes to increase the velocity of the molecules is recorded as pressure, and the greater the number of modes in which this supply of energy is divided up, the less the pressure developed for a given addition of heat. In the case of the monatomic gases, such as argon, helium and the like, the molecules are spheres, which acquire no spin under their mutual collisions. The whole of any heat added to such a gas (at constant volume) is, therefore, effective in raising the pressure, and were a turbine operated with such a gas the reheat factor would have its maximum value, being 1.256 when the pressure ratio is 200 to 1 and the hydraulic efficiency 0.7. In this connection it may be noted that mercury vapor has also spherical monatomic molecules, and Mr. W. L. R. Emmet, of the General Electric Company, described a mercury vapor turbine in a paper read before the American Institution of Electrical Engineers in 1914. If this vapor were superheated the turbine would benefit by an abnormally high reheat factor.

In the case of diatomic gases only three-fifths of any heat energy supplied at constant volume goes in developing pressure, the remainder merely increases the rate at which the molecules spin; whilst with still more complicated molecules still less of any heat energy supplied is effective in generating pressure. In case of water vapor a certain proportion of double molecules are always present, otherwise the reheat factor would be the same as for diatomic gases. These double molecules are not only capable of spinning about three coordinate axes instead of two, but the bond between the components is far from rigid, and hence these double molecules have a low reheat factor. Further, as the temperature rises the number of these double molecules present diminishes, so that some of the heat energy supplied goes in breaking the bonds between the two components. This still further reduces the heat available for increasing pressure, and the result is that the value of γ for superheated steam is 1.3 instead of 1.4, which would be expected were no double molecules present. With wet steam matters are still worse, since much of the heat added goes in evaporating the particles of suspended moisture, a fact which sufficiently accounts for the low reheat factor characteristic of such steam.

Although a considerable proportion of the anomaly in superheat correction is accounted for when the difference in the reheat factors of wet and superheated steam is taken into consideration calculation shows still outstanding discrepancies.

These appear mainly attributable to the fact that the steam passing through a turbine is under-cooled or super-saturated during a substan-

tial proportion of its total range of expansion. Indeed, Professor H. L. Callendar, in a paper contributed to the Institution of Civil Engineers in 1898, gave reason for believing that this condition of affairs also obtained in the cylinders of ordinary reciprocating engines. If this be true, and for our own part we do not question it, the determination of the "missing quantity" at different parts of the stroke of such engines by a study of the indicator diagram is grievously complicated, to say the least, since all methods so far proposed for this determination assume that the steam is in thermal equilibrium throughout the whole of its expansion.

In a turbine the conditions are much more favorable to the phenomenon of under-cooling, and there are strong theoretical grounds for believing that it actually occurs to a very high degree. Direct experimental evidence of the fact is, however, very difficult to secure. This was pointed out by Callendar in the paper already cited. Any thermometer placed in superheated steam indicates, not the temperature of this steam, but that of the film of water which is immediately deposited on the bulb. The temperature of this film tends to be in equilibrium rather with the pressure than with the temperature of the surrounding steam. Hence a very high degree of under-cooling may be represented by a defect in the thermometer reading of but a few degrees. Theory indicates that a high degree of under-cooling does not persist throughout the whole range of expansion, and in ordinary turbine practice the maximum to be expected in the exhaust steam is not more than some ¾ degrees C., although at some previous stage the steam may have been under-cooled some tens of degrees. The higher the initial superheat of the steam the longer is condensation delayed, and with very high superheats, therefore, the steam may be exhausted in a greatly under-cooled condition.

We are indebted to Mr. W. Chilton, of the Brush Electrical Engineering Company, Limited, Loughborough, for the results of some very careful observations made in these conditions. These are reproduced below:

	½-load.	¼-load.	⅛-load.
Exhaust pressure, inches of Hg.....	0.671	0.554	0.569
	deg.	deg.	deg.
Corresponding temperature, F.	67.2	61.6	62.3
Observed temperature, F.	61.0	57.2	60.3

The readings were taken at the turbine exhaust branch, and the conditions precluded the possibility of the low readings being due to conduction losses. For reasons already given the defect of 6 degrees in the trial at half-load probably corresponds to an under-cooling of the exhaust steam by perhaps some 40 degrees or 50 degrees F.

Theory indicates that the loss due to under-cooling is less the longer the phenomenon is delayed. As it can only occur within the saturation field, the effect of superheat is to postpone the production of the supercooled condition to the later stages of the turbine, with a consequent reduction in the under-cooling losses. We hope in a future issue of "Engineering" to set forth in detail the theory, of which the conclusions have been summarized above.—"Engineering."

DESIGN OF OIL-RING BEARINGS.

By William Knight.*

A correct design of bearings is one of the main points to be considered when designing a machine. If they are not suitable for the particular service that the machine is intended to perform, trouble may be expected during its operation, and no matter how well designed all other parts are, the final result will be a failure.

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The function of bearings is to withstand the load imposed on them and to allow for a free rotation of the shaft with the least frictional loss, without undue wear, vibration or oil leakage; this being particularly important in electrical machines and high-speed rotors in general in which vibrations are always objectionable.

It is the aim of this article to describe the main points to be considered when designing a bearing in order to obtain the desired results, chief of which is the supplying of a sufficient amount of oil all along the journal, so as to maintain an unbroken film.

In order to point out the importance of supplying the oil at the right place a résumé is given of the investigations on the points of maximum and minimum pressure in the oil film, made by Tower, Moore and others.

Tower's experiments ("Proc. Inst. Mech. Eng.," 1885) were made on a half-bearing 4 inches diameter and 6 inches long; chord of arc of contact 3.3 inches. Nine holes were drilled in three longitudinal rows, one at the bottom and the other two rows spaced 1 inch from the central one, and located one at the side where the journal entered and the other where the journal left the bearing.

The distance between holes in the same row was 1 inch, each row starting at approximately the center of the bearing and extending toward one end. Each hole was connected to a pressure gauge, the average bearing pressure (total load divided by the projected area of contact) was 311 pounds per square inch and the oil (bath lubrication) was kept at a constant temperature of 90 degrees F. The pressure at the three holes nearest to the middle of the bearing was found to be a maximum at the bottom hole, or twice the calculated average pressure. At the hole on the leaving side of the bearing the pressure was 60 per cent. more than the average, and at the entering side it was 19 per cent. more than the average. Gauge readings at the other holes showed in every row a slight gradual drop in pressure from the middle plane toward the end of the bearing, the pressure at the end holes being an average 87 per cent. of the pressures at the middle of the bearing. All these pressures were found to vary directly as the load on the bearing and to be independent of the journal speed (between 150 and 20 feet per minute). This, however, does not agree with Reynold's theory, which shows that the distribution of the pressure in the film varies with the speed.

Beside having points of maximum and minimum pressure in the oil film, H. F. Moore in 1898 ("American Machinist," Sept. 10, 1903) proved with some tests on a small bearing that at some points of a complete circular bearing, the pressure in the film may even drop below the atmospheric pressure and suck oil into the bearing from a level 6 inches below.

PRESSURE ON OIL FILM.

This point was first proved mathematically by Reynold ("Phil. Trans. Royal Soc. London," 1886). When the bearing is first started from rest, the point of contact between the journal and journal box moves back in a direction opposite to the rotation a small angle, following the law of dry friction. As soon as the oil starts lubricating the rubbing surfaces, the point of maximum approach is gradually shifted about 90 degrees in the direction of the rotation. By increasing the load beyond a certain limit or by decreasing the speed of rotation (load and speed being inversely related in Reynold's equation), the point of nearest approach of the journal and bearing surfaces is shifted back from the horizontal plane (load on the bearing assumed to be acting vertically downward) to 40 degrees from the vertical on the off side of the journal, and the thickness of the oil film at that point is, according to Reynold, equal to one-half the difference of the radii of the journal and journal box.

When the point of nearest approach on the off side is 40 degrees from

the vertical, the pressure in the oil film is everywhere greater than the atmospheric pressure.

By further decreasing the speed or by increasing the load, the point of nearest approach is shifted farther back in a direction opposite to the rotation toward the bottom of the box, the thickness of the film becomes less, and the pressure reaches a maximum value at a point slightly before the point of maximum approach and becomes negative at a point slightly beyond that point, until finally, if the pressure is still further increased a certain amount or else the speed decreased, the oil film breaks at the point of negative pressure.

By increasing the speed, the extreme position of the point of maximum approach in a bearing with the load acting vertically downward is on the horizontal plane at 90 degrees from the vertical in the direction of the rotation. Up to a certain speed the points at which the pressure is respectively a maximum and zero are located very near to the point of maximum approach—the former just before, the latter just beyond. By further increasing the speed, these two points begin to diverge from the horizontal plane for a corresponding increase in thickness of the oil film at the point of nearest approach, with a tendency at very high speed to become diametrically opposite to each other in a vertical plane when the journal is located exactly in the center of the journal box (a position which is not reached in practice).

These investigations on the location of the point of maximum and minimum pressure with reference to the plane in which the load on the bearing acts lead us to the determination of the most desirable point at which the bearing must be supplied with oil.

Evidently the oil must be supplied in that region where the pressure is a minimum, unless forced lubrication is used. In order to find that point, it is necessary to first determine the direction of the resultant of the forces acting on the bearing (gravitation only, or else gravitation and whatever external force is applied to the shaft). The point of minimum pressure in a bearing subject to a moderate pressure and rotating at a speed not too low is in a plane located, in the average case, at about 120 degrees from the plane where the resultant of the forces on the bearing is acting, measured in the direction of the rotation; and the point of maximum pressure is located at about 60 degrees back of it (or 60 degrees from the plane of the resultant, measured in the direction of the rotation), provided that a perfect oil film is maintained all around the shaft.

PROVIDING OIL GROOVES.

If the direction of the resultant was always known, it would be easy to provide oil grooves in the most convenient places, which would allow the oil to be admitted at the right points; but in some cases, as for instance, in electrical motors, the direction of the pull on the shaft due to the belt or to a gear cannot always be anticipated and the bearing must be designed to provide the best lubrication possible under the conditions.

The bearing surface where the pressure is a maximum should be as free as possible from oil grooves, and the first oil-ring lubricated bearing, used in 1860 by Benjamin Hick & Son, Bolton, England, was a failure because the oil grooves were cut in the lower half of the bearing instead of in the upper half, where they would have been effective.

Oil grooves should be designed, first, to conduct the oil to the center of the bearing from where it spreads out toward the ends; second, to provide on the bearing surface a reservoir of oil which will always maintain a generous supply of oil throughout the length of the bearing; and third, to catch the oil toward the ends and return it to the previously mentioned reservoir, thus preventing, as much as possible, the escape of the oil from the journal box.

The direction of the rotation should also be considered in designing the oil grooves for a given bearing, because if they are not cut in the proper direction, they will not assist in carrying the oil from the center supply to the ends as intended. In electrical motors designed for right- and left-hand rotation it is necessary to strike a happy medium in providing grooves, which are a necessary evil in bearing design, and they must be cut symmetrically for both directions of rotation. Too many grooves will unnecessarily reduce the bearing surface and will assist in breaking the film. Improper grooving will have the same effect.

Having provided for the distribution of the oil over the surface of the bearing where the pressure is a minimum, provision must be made so that it is not wiped off and is allowed to reach the surfaces more heavily loaded. This is provided for in bearings loaded on the bottom half with some secondary grooves (chamfers) cut lengthwise at the split of the bottom half of the journal box and extending to within a short distance from the ends, or by counter-boring the journal box at the split in such a way as to allow the oil to collect there and to be drawn into the lower half between the journal and journal box over the length of the bearing.

In general, no oil grooves should be provided in the lower half of a bearing with the load acting on the bottom half, except two shallow grooves extending around the lower half of the journal box (in large sizes only) near the ends, in which the oil that is squeezed out can collect and be again carried into the chamfer or the counterbored recess at the split on the other side.

Grooves and chamfers it is generally assumed should be cut smoothly and have no sharp edges which might scrape off the lubricant. That sharp edges are detrimental to the maintenance of a perfect film is rather questionable, and I know of an engineer who has proved with numerous tests the fallacy of the opinion generally held by manufacturers and users of journal boxes in regard to this point.

WIDTH AND DEPTH OF OIL GROOVES.

The width and depth of oil grooves is another matter to be considered. They should be wide and deep enough to insure a sufficient supply of oil to be distributed all over the bearing surface.

The size of the bearings governs the size of the oil grooves, but in most cases from ⅛ to ¼ inch width and 1/16 to 5/32 inch depth are sufficient. Besides accomplishing this main function, the oil grooves will be useful if metal rubbing should occur with a consequent rubbing off of the bearing metal. In that case the abraded metal will collect into the grooves and fill them in before seizing occurs, and the machine can be frequently stopped in time to avoid serious damage to the bearing if the heating is noticed in time.

When a bearing shows a tendency to heat, it is well to inspect the oil grooves, because they may have become partly obstructed by the babbitt or other material and may prevent the flow of the oil.

J. C. K. Balfry, in a paper read before the Rugby Engineering Society, states that many firms in England are designing high-speed bearings without any grooves at all and they believe it to be sufficient to supply oil along the whole length of the bearing at about the split. The theory of the high-speed bearing developed by Reynold shows that grooves in such a bearing do not serve any purpose, and it may even be argued that they are actually detrimental.

It is a maxim that similar metals must not be used for journal and bearing, except in the case of hardened steel and cast iron. This is probably so because they are of the same hardness, and it is conceivable that crystals of the same hardness have some sort of affinity and com-

bine together under favorable conditions by alloying or welding, more easily than crystals of different hardness.

Low-carbon steel journals are likely to give trouble where high-carbon steel journals would give none, and this fact also shows that in the case of low-carbon steel a greater affinity is likely to be found between the journal and some crystals which may become detached from the bearing metal.

Gray unchilled cast iron, owing to the presence of graphite and to its porosity, makes excellent bearing surfaces when working in contact with steel or wrought iron, and also against itself. It does not seize readily and the only objection to its use as a bearing metal is its brittleness.

BEARING METALS.

For electrical machines two classes of bearing metals are generally used—bronzes and white metals. The fundamental difference between bronze and white "anti-frictional alloys" is to be found in their microscopic structure. In bronzes plastic crystals of copper are embedded in a hard eutectic, and if, by reason of defective lubrication, metal contact should occur at any spot between the journal and the bearing surface, particles of copper will separate from the mass and become welded or alloyed to the journal (especially if this is made of soft steel), forming a rough surface which will attract at every revolution fresh particles of copper until serious heating and damage result.

White metals instead are composed of hard grains having a low coefficient of friction (which for this reason do not easily cut the journal) embedded in a plastic alloy.

Bronzes are alloys composed chiefly of copper and tin. The presence of tin confers strength to the alloy, which up to about 13 per cent. of tin is a tough homogeneous solid solution of tin in copper. If the percentage of tin is increased beyond this limit, the alloy becomes hard and brittle and increases in hardness and brittleness in proportion to the increased percentage of tin.

COMPOSITION OF BEARING METALS.

Phosphor bronze is an alloy of copper, tin and phosphorus. When used in the proportion of from 84.5 to 91.0 per cent. of copper and 0.37 to 0.85 per cent. of phosphorus (see "Journal Inst. Metals," page 164, 1909), a net-work of hard copper-phosphide embedded in a soft matrix of copper-tin alloy is formed and a metal particularly suitable for bearing service will result, with a low coefficient of friction. According to Philip such an alloy is particularly suitable for high-speed machines.

The addition of lead (provided that the metal is well stirred up before pouring and is not poured at a too high temperature, so as to insure a good diffusion of the lead throughout the molten mass) greatly increases the plasticity and reduces the wear. An alloy of 79.7 per cent. of copper, 10 per cent. of tin, 9.5 per cent. of lead and 0.8 per cent. of phosphorus was proved by Dudley's experiments ("Journal Franklin Inst.," 1892, pp. 81-93 and 161-72) to be a metal sufficiently strong, plastic, of good wearing properties and of a finely granular structure so as to make it particularly suitable for bearing metal. The addition of lead to bronze alloys was the first step toward the adoption of plastic white-metal linings in rigid-bronze, cast-iron or steel journal boxes.

Lead is a cheap metal, but is too plastic to be used alone and must be hardened by adding antimony and tin. Babbitt metals were eventually introduced and are now largely used. The original proportions used by Babbitt as given in Alford's book, "Bearings and Their Lubrication," were:

Tin = 89.3 or 83.3 or 89.1
Copper = 3.6 or 8.3 or 3.7
Antimony = 7.1 or 8.4 or 7.2

White metal and babbitt alloys are particularly suited for the lining of bearing shells owing to their plastic nature, which enables them to mold themselves to the form of the journal, thus providing a larger bearing surface with a more even distribution of the load. High spots can be more easily scraped down, and a smooth and polished bearing surface is obtained. Bearing surfaces of white metal and babbitt are also more economical than bronze, because of the comparatively low cost at which a new lining can be provided in a worn-down or otherwise damaged journal box.

White metals are alloys of tin, antimony, lead, copper and zinc. Other elements, like iron, bismuth, aluminum and graphite, enter in the composition of the alloy, and as for bronzes, the different proportions into which the constituent parts are alloyed together will provide a bearing metal more or less suitable for a given service. For instance, for bearings subjected to heavy pressure or severe shocks a hard mixture is required; a soft one would be rubbed off in flakes. An alloy of 80 parts of lead, 12 parts of tin, 7.5 parts of antimony and 0.5 part of copper is a good bearing metal for heavy loads. An alloy composed of 80 parts of lead and 20 parts of tin is particularly suited for high-speed bearings and moderate loads. A Brinell hardness of 23.5 for lead-base and 30 for tin-base babbitts have been found to give excellent results.

CAUSE OF HOT BEARINGS.

As already pointed out, white-metal alloys are made of hard grains embedded in a plastic matrix, and as for bronzes, a coarse size of grains is to be avoided in order to prevent damage to the shaft.

Babbitt linings must be provided with a sufficient number of anchoring buttons to secure a firm connection with the cast-iron shell. This is an important point in bearing construction, and the cause of many hot bearings can be traced back to a lack of contact between the bearing metal and the outside shell.

This lack of contact may be produced by the shrinking of the babbitt when poured in the journal box, due to the difference in the coefficient of expansion and temperature of the babbitt and the bearing shell (the coefficient of expansion of the babbitt being two or three times larger than that of cast iron). When secured with buttons to the outer shell, the buttons will prevent the babbitt from becoming loose, but for this very reason shrinkage stresses are produced in the babbitt, which under shocks and heavy loads will develop cracks that will become gradually impregnated with oil. The presence of an oil film between the babbitt and the outside shell at those places where a good contact is not obtained, owing to the very low heat conductivity of the oil, will prevent the dissipation of the heat from the babbitt to the shell and from this to the atmosphere, thus increasing the temperature of the babbitt.

To correct these troubles as much as possible, an intelligent choice should be made of the most suitable alloy to be used for a given service, and proper consideration should be given to its physical characteristics (tensile and compressive strengths, elongation, reduction of area and hardness). A sufficient number of buttons should be provided for, and great care should be taken when pouring the babbitt that no dross or any other foreign substance is present in the liquid mass of metal, and over-heating of it should be carefully avoided.

The practice of one of the largest electrical machine manufacturing

concerns is to pour hard babbitt at an average temperature of 775 degrees F. and soft babbitt at an average of 920 degrees F., allowing not more than 115 degrees F. variation both ways. Overheating is more serious in soft babbitt than in hard. The shell should be heated to between 210 and 300 degrees F. before pouring in the babbitt, and babbitted bearings should not be jarred during the solidification of the metal.

In order to obtain as good a contact as possible between the lining and the shell in a journal box, the bearing-metal lining must be carefully peened. An ideal metal would be one expanding during the process of solidification instead of contracting, and in this class are antimony alloys.

BEARING CLEARANCE.

Between the journal and the journal box a sufficient clearance must be provided to insure a good film lubrication and to take care of the expansion of the babbitt. The coefficient of expansion of the babbitt being two or three times larger than that of either cast iron or steel and the babbitt being prevented by the shell from expanding outwardly, the expansion will have to take place inwardly, and if a sufficient clearance is not provided for, the oil film may be broken and binding may occur. A clearance of 0.001 inch per inch of bore plus 0.002 inch will be about right for electrical motors and generators, provided that the deflection of the shaft inside of the journal box, due to the weight of the rotor, the belt pull or an overhanging load, is less than the clearance between the journal and the journal box.

Large outfits are generally equipped with spherically seated journal boxes, which will insure a better alignment of the shaft and a more uniform thickness of the oil film due to a better equalization of the load over the entire length of the bearing. With self-aligning bearings, a clearance of 0.001 inch per inch of bore plus 0.002 inch will do.

OIL-RING BEARINGS.

With oil-ring lubricated bearings another point to be considered is the design of the oil ring. In order to have efficient lubrication, these rings must be properly designed and constructed. It is generally assumed that an oil ring cannot be expected to distribute oil over a width of bearing of more than 8 inches (or 4 inches at each side of it), so that the maximum length of bearing with which a single ring at the center could be used is 8 inches; for bearings between 16 and 8 inches long two rings should be used located at a distance from the ends equal to about one-quarter the bearing length; for a bearing of 24 inches long, three rings should be used, and so on. As a matter of fact, however, the writer knows of bearings 27 inches long where two rings only are sufficient to bring up an amount of oil sufficient to insure good lubrication.

Good proportions for oil rings are obtained by making them about twice the diameter of the journal and providing an oil level in the oil well at a distance from the shaft about equal to half the diameter of the journal.

All corners and sharp edges in the ring should be avoided, and in split rings there should be a good contact between the two halves at the joint, so that the oil cannot squirt out through the split. Rings should be made perfectly round and well-balanced in order to prevent "dancing" of the ring on the shaft. When dancing occurs, the ring stops and no oil is brought up to the journal. This gradually increases the friction between ring and journal until the ring, because of the in-

creased friction, starts rotating again, brings up some oil, then stops and starts dancing again.

PREVENTING OIL THROWING.

Owing to defective working of the oil rings, oil is sometimes thrown against the bearing cap and, owing to capillary action, creeps out through the split. Oil leakage at the split is to be particularly avoided with electrical machines on account of the destructive effect of oil upon the insulation of the coils and commutator. This can be prevented with a groove at the split of the pedestal which will drain the oil reaching there into the oil well again without allowing it to leak out, or else an oil guard may be cast in the bearing cap, extending below the split and inclosing the oil ring.

Another place where oil troubles may be anticipated and efficiently prevented with a good design is at the shoulder of the journal. At this place, owing to the end play of the shaft, the oil is squeezed out between the shoulder on the shaft and the end of the journal box. To prevent the oil from being thrown out of the bearing, a groove is cut in the journal box either into the bearing surface adjacent to the thrust shoulder or else outside of it, opposite to the oil sling, with drain holes at the bottom which will allow the return of the oil into the oil well. If the latter construction is used, a small clearance should be provided between the journal and the journal box at the oil sling, such as to insure a sort of sealing effect, and the oil sling on the shaft should be made wide enough to prevent the lip of the journal box from ever clearing the shaft and thus destroying the desired sealing effect, owing to the end play.

Adjacent to the oil sling an oil groove is provided on the shaft, so that whatever oil may happen to leak out of the journal box over the oil sling is driven back again by centrifugal force without being allowed to get out of the bearing. In some designs, adjacent to this groove another oil sling similar to the one adjacent to the thrust shoulder is cut into the shaft. Opposite to these two slings a second oil groove could be made in the journal box, similar to that previously described and accomplishing the same function. In this way the oil, being squeezed out between the thrust shoulder and journal box, is forced to go in a first chamber in the journal box having drain holes at the bottom.

If the amount of oil being squeezed out into that chamber is too much, the excess which does not flow through the drain holes will leak out and will fall in the groove on the shaft, whence it will be thrown back again, a part going back to the first chamber and another part filling the outer chamber in the journal box.

The same effect could be obtained by having a thrower mounted on the shaft near the thrust shoulder surrounded by the bearing housing, which would catch the oil thrown out of the bearing in a direction parallel to the axis of the shaft.

Dust should not be allowed in the bearing. This may be provided for by using a felt or leather washer fastened to the bearing housing and wiping on the shaft.

Another important point to be considered is a rigid construction of the pedestals or shields supporting the bearings and a good alignment of the shaft. A springy support might cause trouble, and a bad alignment of the shaft will offset all the best care taken in properly designing bearings.—"Power."

AN OUTLINE OF THE THEORY OF HEAT TRANSMISSION AND ABSORPTION IN APPLICATION TO ORDINARY ENGINEERING PROBLEMS.

By John E. Bell.*

Heat can be transferred between substances in three different ways, by radiation, conduction and convection, and it will be my object in this paper to give briefly some account of the theory applicable in each case, in so far as it is useful in understanding the derivation of formulae. Within the limits assigned by the paper it is, of course, impossible to treat any one phase of the subject thoroughly, but it seems to me that much of the important information can be given in a condensed way without losing value. Before entering into the technical treatment I will, with your permission, give some account of the structure of matter as it is now understood, as it may help to make clear statements made later on and, besides, may be of general interest.

Matter is known to be made up of almost inconceivably small particles called molecules which, in different substances, vary in weight and complexity of formation. When the molecules are separated by distances comparable to their size, there is an inter-action which seems to be an attraction when they are near together and a repulsion at a distance. In a solid the attractive forces bind the molecules, so the only motion possible is one of vibration about a position of equilibrium. In a liquid the attractive forces are less and the molecules can move from one position of equilibrium to another. In a gas the molecules are separated beyond the range of attraction and consequently move in straight lines until two collide, when they are deflected in their line of motion by forces which seem to be repellent. The molecules, or rather the atoms which combined form the molecules, were once supposed to be infinitely small particles of matter obeying the ordinary laws of motion. It is now thought, however, that they are of a nebulous structure. At the center is a core or nucleus, and moving around it are numbers of small so-called electrons or corpuscles. These electrons carry a charge of negative electricity and the nucleus contains a balancing charge of positive electricity. The electrons are of one sort in all kinds of atoms. They have been isolated and their mass determined. It is 1/1700 that of an hydrogen atom, and is apparently not mechanical mass at all, as ordinarily understood, but is the inertia effect due to the electro-magnetic field produced by the charge. Part of the electrons are free to move under the attractions of neighboring molecules and pass from one to the other, or else move freely in space. Particularly is this true in good electrical conductors. It also appears that, to a certain extent, in all bodies the molecules dissociate so that any substance is, as a matter of fact, made up of molecules, atoms and electrons. In a solid, as pointed out above, the molecules, while free to vibrate, are held in fixed positions. The electrons, however, move about among the molecules much as the molecules themselves move in a gas, and they carry energy from one position to another.

The velocity of a moving molecule or electron due to collisions and attractions of neighboring molecules is continually changing. Some will have a very little motion, while others will either vibrate or move at a high velocity. There must, however, be some definite distribution of velocities amongst the myriad of particles constituting an appreciable weight of matter, which always keeps a fixed number with velocities between set limits. This distribution of velocity, it is found, makes the

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mean kinetic energy of the electrons the same as the mean kinetic energy of the molecules. This mean kinetic energy of motion of the particles is the same for all substances at the same temperature. It has been known for a long time that this was true of gases, but only in recent years has it been demonstrated to be true for all substances and also for electrons.

When there is a difference in temperature, the molecules and electrons moving in the direction of the lower temperature have a higher velocity and carry more energy than those moving in the other direction. There will, therefore, be a transfer of energy in the form of heat and the process described is that of conduction. In metals the heat carriers are largely the electrons and, for this reason, good electrical conductors are also good heat conductors. In a gas the carriers are the molecules. As the electrons have the same kinetic energy as the molecules at the same temperature, but move with very much higher velocities, it is obvious that conduction in this way will be at a more rapid rate in solids than in gases. The molecules are also closer together and collide more frequently.

When two molecules collide the shock sets the electrons vibrating, and the vibrations start trains of electro-magnetic waves. The energy transferred in this way is said to be transferred by radiation. When the molecules are near the surface of a solid this radiant energy passes out of the solid entirely; in the interior it is absorbed by other molecules and, when there is a difference in temperature, assists in conduction of heat. Radiation from solids is made up of waves of varying lengths, and the proportion of energy carried by waves of a given length varies with the temperature and the physical properties of the solid. If the radiation is from different sources of varying temperatures, the distribution of energy between different wave lengths might be most anything desired. It is true that the stream would carry a certain amount of energy, but this energy could not be associated with any fixed temperature.

The radiation through an opening in an enclosure at a uniform temperature is independent of the substances forming the walls of the enclosure. Radiation from a source of this sort is said to be full radiation. At high temperatures all solids seem to emit full radiation, but this is not even approximately true at lower temperatures. It is for this reason that radiation pyrometers, which measure only the total amount of energy received, to read accurately should be exposed to full radiations from surfaces all at the same temperature. If the radiation is from surfaces of different temperatures, the reading shows that the sum total of the energy received is the same as that from a source giving full or black body radiations at the temperature indicated.

A body emitting full radiation loses energy at a rate proportional jointly to the bounding surface and to the fourth power of the absolute temperature. If the surface is such that one part can radiate heat to another part, this statement is not accurately true. For the total energy emitted to be proportional to the surface the rays must travel clear from every point. This law is known as Stefan's law and was originally stated as an empirical expression of experimental results that were later on shown to be in error. Its accuracy, however, has since been established experimentally, and Boltzmann has deduced it entirely from the theory of electro-magnetic radiation so that it is now considered a statement of fact. Expressed in B.t.u.'s per hour per square foot of surface, the radiation from a body at the temperature T in degrees Fahrenheit (absolute) is very nearly

$$1{,}600\left(\frac{T}{1{,}000}\right)^4.$$

Very hot bodies, regardless of their physical characteristics, seem to emit heat at the rate just mentioned. At lower temperatures the physical characteristics of the surface influence the wave lengths of the vibrations, and the radiation is no longer of the character assumed in the statement of Stefan's law. Inaccuracies in deduction from its practical application are not due to this, however, as much as to inequality of temperatures. In most cases, in addition to the hot surfaces, cooler surfaces are present and absorb radiant heat. These surfaces in turn radiate at a rate dependent on their temperature, and the net effect is measured as the difference between the amount emitted by the hot bodies, according to the law above, and that returned by the cooler surfaces. If the exchange of radiation is between two plane surfaces, one at the temperature T and the other at the temperature t, the net effect is a transfer of heat per square foot of surface of

$$1{,}600\left[\left(\frac{T}{1{,}000}\right)^4-\left(\frac{t}{1{,}000}\right)^4\right].$$

If the higher temperature is 3,000 degrees (2,540 degrees F.) and the lower temperature is 1,000 degrees (640 degrees F.), the heat transferred, according to this formula, is 98,000 B.t.u.'s per hour per square foot of surface, which is heat enough to evaporate 100 pounds of water per hour. If the temperature of the cooler surface is reduced to 500 degrees (40 degrees F.), the heat transferred would be almost the same, so that when there is as much difference in the temperatures as that assumed, the actual temperature of the cooler surface is of little consequence. In very few cases is the exchange between surfaces as simple as between two parallel planes emitting and absorbing full radiations. In a boiler, for instance, the tubes exposed to the furnace heat receive radiation from the very hot fire bed and from surfaces of the brick wall that vary in temperature from something near to that of the fire bed to a temperature not much different from that of the tube. Under these circumstances, it is not possible to make any accurate calculation as to how much heat can be exchanged by radiation, even though it is assumed that Stefan's law applies. Nevertheless, if it is conceived that the radiation is from a glowing surface, at the temperature a pyrometer would show in the furnace, the radiant heat absorbed is, I believe, not much different from that given by Stefan's law. I have had a number of opportunities of checking this roughly. In one case a lower tube was removed from a B. & W. boiler fired by a Taylor stoker. The tube was replaced with another shorter one of the same size, which extended through the header at one end and just entered the first baffle at the other. Both ends were closed and connections made for circulating water through it. The water was weighed and the rise in temperature taken. The heat absorbed corresponded to an evaporation of 75 pounds per square foot of surface per hour, more than ten times the average for the boiler. The surface on the top of the tube did not receive radiation, and the bottom surface not only received radiation, but also the heat due to the impingement of the furnace gases, which, I imagine, is at a very high rate. If two-thirds of the surface is considered to be effective, the evaporation would correspond to 112 pounds per square foot. Of this, more than likely, 100 pounds was from the radiation and 12 pounds from the heat of the first impingement of the gases. A pyrometer in the furnace would have recorded about 2,500 degrees F., so that the evaporation observed checked with the formula with as much accuracy as can be expected. More recently the Power Specialty Company experimented with a new design of superheater placed in the roof of a furnace, directly over the grate surface, which received its heat entirely by radiation.

The heat absorbed was found to be little different from that predicted by the formula.

The amount of heat transferred by radiation falls very rapidly with the temperature. A drop in temperature from 2,500 degree F. to 2,400 degrees F. will reduce the amount so transferred by 12 per cent. It rises with temperature still more rapidly. It is for this reason that boilers with hot furnaces can so easily be overloaded.

If, to refer to the formula again, the higher temperature is 120 degrees F. and the lower temperature 60 degrees F., the heat transferred is 64 B.t.u.'s, or at a rate of 1.07 B.t.u.'s per hour per square foot of surface per degree Fahrenheit difference in temperature. Experiments with radiation in a vacuum at these temperatures give higher results. McFarlane's experiments, referred to in Preston's "Theory of Heat," give 1.7 B.t.u.'s for polished surfaces and 2.4 B.t.u.'s for blackened surfaces. It is probably not far out of the way to assume that heat is lost by radiation from bodies at ordinary temperatures at a rate of 2 B.t.u.'s per square foot of surface per hour per degree Fahrenheit difference in temperature.

If a pyrometer with a protecting casing is used to measure temperatures, the effect of this radiation should be considered. If, for instance, the pyrometer is put in between the tubes of a boiler, it will read low. How much the reading will be off will depend on the rate of absorption of heat by contact with the gases, as compared with the radiation loss. With a smoky gas an interchange of radiations is with particles at about the gas temperature, so there should be no correction to make. With a clean gas, however, the radiation is to the boiler surface. If the letter n is used to represent the ratio between the rate of transfer due to radiation and rate of transfer due to exposure of the gas flow, the pyrometer rating must be increased by n times the difference between the pyrometer reading and the temperature of the boiler tubes. The value of n will depend on the velocity of the gases; usually it will be probably about 0.25. Where the difference in temperature is 100 degrees the correction will be 25 degrees. If, however, the difference in temperature is as much as 1,000 degrees, the correction would be around 250 degrees.

If, when considering radiation between two plane surfaces, an intermediate surface is included, this must receive as much heat from the hot side as it radiates from the cold side, and in each case the amount of heat is equal to that transmitted from the hotter surface to the colder surface. If the equations according to Stefan's law are written down for both sides of this plate and added together, the temperature of the plate is eliminated and the result is that the total transfer is in accordance with Stefan's law, with a constant of just one-half the amount. If two intermediate surfaces are in position, the radiation constant is reduced to one-third, and so on. It is evident that such an arrangement would constitute an effective screen against the losses of heat by the hotter body. A pyrometer inside of a cylinder with the gas flowing freely on both sides will not be affected in its reading by the temperature of the surrounding bodies anything like as much as it would without the screen. A plate set along the side walls of a boiler a few inches away from the surface will not be as hot as the surface, and if the air for combustion is drawn down between the furnace wall and this plate the insulation against a loss of heat is probably as effective as can be obtained. In a gas the interchange of radiation between molecules is somewhat in this same way, so that its effect on the conductivity is limited.

It was pointed out in the foregoing that heat was conducted through gases by the vibrating molecules and through metals, to a large extent, by vibrating electrons. In other solids which are non-conductors of electricity and in liquids the heat is probably transmitted largely by contiguous molecules clashing against each other in their vibration, and

possibly by inter-molecular radiation. The electrons, on account of their high velocity, should be the most effective as carriers, and it is therefore to be expected that the metals show the highest conductivity. Other solids and liquids should have values much lower, and this experimental data shows to be the case.

It is convenient to refer to conductivity of substances as measured by the number of B.t.u.'s transferred per hour through a slab of the material one foot square and one inch thick when the difference in the temperature of the two faces is 1 degree F. This unit is 2,904 times the C. G. S. unit used in scientific tables. The values of this coefficient for a number of substances I give below, with a note of the authority:

Aluminum	1394	at 64° F.	(Smithsonian Physical Tables)
Copper	3666	at 64° F.	(Smithsonian Physical Tables)
Cast Iron	314	at 64° F.	(Smithsonian Physical Tables)
Wrought Iron	418	at 64° F.	(Smithsonian Physical Tables)
Soft Steel	322		(Smithsonian Physical Tables)
Hard Steel	180		(Smithsonian Physical Tables)
Water	4.16	at 68° F.	(Encyclopaedia Britannica)
Fire Brick	4.5		(A mean value only)
Air	0.165	at 60° F.	(Dynamical Theory of Gases—Jeans)
Hydrogen	1.6	at 60° F.	(Dynamical Theory of Gases—Jeans)

The conductivity of perfect vacuum would be zero. Heat is only transmitted through a vacuum by radiation.

The conductivity of all substances vary with the temperature in a manner that is not entirely understood. The electronic theory of conduction in metals makes the ratio between the heat conductivity and the electrical conductivity proportional to the absolute temperature, and in good conductors this seems to be the case. The thermal conductivity on this same basis would be constant, but as the theory has not, as yet, been satisfactorily worked out in all details, this undoubtedly is incorrect. There is hardly enough data for liquids to say whether the value increases or diminishes with a rise in temperature. In gases the heat carriers, the molecules, also carry momentum from one layer of gas to another when subject to a shearing motion and, according to the kinetic theory of gases the coefficient of conduction is directly proportional to the specific heat at constant volume times the coefficient of viscosity. This relation seems to hold very well for the permanent gases. If it does hold, the coefficient of conductivity for air has a temperature factor:

$$\frac{C_v T^{\frac{1}{2}}}{T + 185}$$

where T is the absolute temperature in Fahrenheit degrees.

The high value of the coefficient for hydrogen is due, of course, to the high specific heat of this gas.

The change of the coefficient with pressure in a gas is probably negligible for all ordinary pressures, as it is known that the coefficient of viscosity is independent of the pressure.

It is almost impossible to measure the conductivity of either liquids or gases accurately on account of the convective currents due to differences in density. It is, for this reason, that little is actually known about the variation with temperature. It would seem, however, that the actual value of these coefficients is relatively unimportant in the theory of heat transfer as, in this particular, the predominating influence is the internal motion of the fluid.

The motion of fluids has always been a subject of interest to physicists and mathematicians and has received a great deal of attention.

In spite of this, the electronic theory which has been developed in the last fifteen years lends itself better to the interpretation of experimental data than any general theory of fluid motion so far propounded. This is entirely due to the impossibility of solving the general equation for the motion of a fluid subject to internal friction due to what is known as viscosity. The only motion subject to treatment is that used in determining the coefficient of viscosity of the fluid, namely, a straight stream line motion through a channel, or a motion in circular stream lines between rotating cylinders. In both of these cases the velocity of a fluid particle does not change during its motion. When the velocity does change, as is true in actual cases, the mathematician is entirely helpless. The most complete treatment of this subject in any language is Lamb's "Hydrodynamics." There is a new edition out this year. Lamb does not give a solution of even the simplest problem of the motion of a viscous fluid when the velocity varies throughout a stream line.

In a so-called perfect fluid it is assumed that there is no resistance to a shearing motion, and the theory on this basis is quite complete. It is sufficient to account for the wave motion in bodies of water and for sound waves in gases. Also, to a lesser degree, the formulae for jet motion in two dimensions for flat plates seem to check up with the experimental data, but this is not true for cylinders or spheres moving through a fluid. Even assuming the absence of friction, there is no accurate formula applicable to the motion of a gas where there are variations in pressure and density. I have elaborated a little on this point as, unless one is familiar with the subject, he is apt to be misled by much that has been published in connection with heat transfer formulae into believing that it is possible to calculate results from a fundamental basis.

The theory of the motion of a perfect fluid contained between surfaces requires that the fluid move without friction and with assignable velocities over the surfaces. Since fluids are known to be of a granular construction and solids are also known to be of the same nature, it is a physical impossibility for any fluid to have a velocity next to a surface other than the velocity of the surface itself. In gases at very low pressure the fluid characteristics are lost, and it has been customary to refer to a slip at the walls, but this does not in any way affect the statement just made. There are many reasons, however, for believing that at some distance from a boundary a fluid will move as a perfect fluid, as in observed motion the energy lost by friction is small in comparison with the total energy. Again, smoke rings and other forms of vortex motion follow very accurately the predictions from the theory. In circular pipes, however, and other forms of channels this is not the case at low velocity. Professor Osborn Reynolds has popularized this knowledge by experiments with glass tubes, through which water flows, with a small colored stream at the center. At low velocities this was drawn out to a fine ribbon. As the velocity increased a point was reached where the colored stream would wave from side to side and would then break up and the color would be diffused from wall to wall, indicating the presence of cross currents comparable in magnitude to the mean velocity through the tube.

In all formulae that are accurate and not empirical every term must be measurable in the same units, whatever this unit may be. When the number of quantities that can enter into the result is limited to three, four or five a great deal of knowledge has been derived from this one consideration. Professor Reynolds assumed that motion in a pipe was influenced only by the mean velocity of flow, the viscosity, density and diameter of the pipe. On this basis he concluded that the critical velocity in all fluids would be proportional to the coefficient of viscosity divided by the product of the density and diameter, and experiments in a way

check this out. If, however, vibrations such as sound waves have an influence on breaking down the motion, this conclusion would not be correct, and recent investigators seem to believe that something of this sort actually occurs. When the pressure drop is included there are five quantities involved in the motion. These can only be combined in two ways to give terms of zero dimension, and if it is assumed that these five measurable quantities are the only ones to be considered, one of these terms must be a function of the other. Stanton and Pannell, following out a suggestion of Lord Rayleigh, have shown that this is true, and from their experiments formulae for friction loss of pressure have been developed that are applicable to any fluid, provided the motion is one that does not change the density. In this formula the coefficient of viscosity produces a negligible result at high velocities. The constant in the ordinary pressure drop formula at high velocity holds for all fluids. It is for this reason that I made the statement above that the coefficient of conductivity, which is directly related to the coefficient of viscosity, has probably little bearing on heat transfer in pipes or channels. Heat is carried across the pipes to the walls by cross currents.

If in a pipe the fluid is supposed to carry a series of diaphragms or pistons that move with the mean velocity of the flow, it is obvious that there will be a rolling motion from side to side, something of the character I have sketched out in Fig. 1. In effect the motion is much the

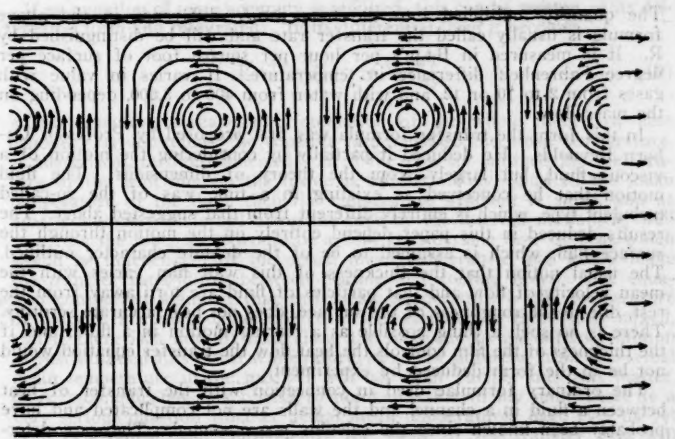

FIG. 1.—STREAM LINES OF A FLUID IN A PIPE RELATIVE TO AN OBSERVER MOVING WITH THE MEAN VELOCITY OF THE FLUID.

same as if there were a closely packed procession of vortex rings just like the ordinary smoke rings. The mean velocity in this motion has a very specific meaning, and the observed velocities across the section of the pipe would vary much, as they actually do. Along the walls there must be a thin layer, in which the velocity increases very rapidly from zero at the walls to something like 70 per cent. of the mean velocity a very short distance away. These two forms of motion, it would not seem, could come together along any surface that would be a surface of equilibrium, and I am inclined to think that it is being continually broken up and eddies or small vortex filaments of the hot fluid are being carried into the layer, where the vortex motion is quickly sheared out. This

form of motion could be determined from the density and viscosity theoretically without introducing any other of the physical characteristics, and this seems to be necessary from Stanton's and Pannell's experiments. The heat transferred through this film would be divided in two parts, a small amount proportional to the coefficient of conduction and a large amount carried by the fluid actually entering the film. The change in momentum of this would also produce a drop in pressure and, if this is taken as proportional to the square of the velocity and the density jointly, the amount of heat carried would be measured by the continued product of the specific heat, density, mean velocity and temperature difference. The product of the density and mean velocity is the weight or mass flow per square foot of channel area, so that the heat absorbed can be written as

$$H = (a + bw)(T - t)$$

Where H is the B.t.u.'s transferred per hour per square foot of tube surface.
T = the mean temperature of the fluid.
t = the temperature of the tube wall.
a = a constant probably proportionsl to the conductivity.
b = a constant that is proportional to the specific heat and also varies with the temperature in some way that so far seems obscure.
w = the mass flow per unit area of the channel.

The quantity multiplied into the temperature difference in the above formula is usually called the transfer rate and will be distinguished by R. It is measured in B.t.u.'s per hour per square foot of surface per degree Fahrenheit difference in temperature. It varies in value with gases from 2 to 10 or 12, and with water from 400 to 3,000, depending on the mass flow.

In this form the transfer formula was first proposed by Professor Osborn Reynolds. He deduced it partially by considering the motion of a viscous fluid, but largely from the theory of dimensions. The fluid motion that he conceived as existing in a tube was of the so-called turbulent type, which is entirely different from that suggested above. The results deduced in this paper depend entirely on the motion through the surface film, which is assumed to be of the definite character outlined. The usual notion that the thickness of this wall film varies with the mean velocity of flow and that particles of fluid are torn away from the rest, due to the roughness of the surface, does not bear accurate scrutiny. There is no such a thing possible as a tearing motion in a fluid, and if the thickness of the film controls the heat flow the transfer equation would not be in the form deduced by experiment.

The ordinary formulae used in connection with the transfer of heat between a fluid in a channel and the walls are not complicated and have probably been known since the calculus was invented. They are determined from equating the heat absorbed by the tube walls, using the transfer equation given above, to the heat lost by the fluid. As stated, this assumes the fluid to be hotter than the tube walls. When, however, the tube walls are hotter than the fluid the same equations apply with slight changes.

The formula most commonly used assumes a transfer rate which applies to the difference between the mean gas temperature and the mean tube temperature, and is usually written as follows:

Representing the temperatures by
T_1 at the hot end.
T_2 at the cold end.
t_1 is the wall temperature at the hot end.
t_2 is the wall temperature at the cold end.

The mean gas temperature is

$$\frac{1}{2}(T_1 + T_2)$$

and the mean tube temperature is

$$\frac{1}{2}(t_1 + t_2)$$

The surface between is S, and the heat given out is equal to

$$H = SR\left(\frac{T_1 + T_2}{2} - \frac{t_1 + t_2}{2}\right).$$

It is also equal to

$$H = Wc(T_1 - T_2)$$

where W is the weight flow in pounds per hour and c is the specific heat. When these two values are equated the first general equation used in heat transfer is obtained

$$S = 2\frac{Wc}{R}\frac{(T_1 - T_2)}{(T_1 - t_1) + (T_2 - t_2)} \quad \ldots\ldots(1)$$

If an equation of more accuracy is required, this can be written in its differential form and integrated. In its differential form it becomes

$$R(T - t) = -Wc\frac{dT}{ds} \quad \ldots\ldots(2)$$

This equation is really the foundation of the whole theory of heat transfer. It determines the transfer rate and also the mean temperature, as these two are interconnected. If in the flow as outlined the temperature of the wall film on the inside is approximately equal to the mean temperature of the fluid, then the temperature gradient is small, and as heat would be transferred along the tube by the internal motion, as well as across the tube, there would be a very considerable flow of heat which would be independent of the actual mean temperature and would make the transfer rates differ between heating and cooling flows. The equation considered assumed that this mean temperature is the temperature of the gas if it could be abstracted at any section and thoroughly mixed, and if there is a flow of heat along the tube due to the internal motion as just outlined this mean temperature would be different from the actual mean temperature in the tube. I will not enter further into this matter, as it is complicated, but I wish to point out that in my mind there is a very serious question whether actual mean temperatures have ever as yet been determined, and consequently the value of the transfer rate will show more or less anomalous variations on this account.

The transfer rate varies some with the temperature. The variation, however, is not marked in most cases. Even where it does vary, if the rate corresponding to the mean temperature is used there is practically no error in considering R as a constant. With this understanding the transfer equation can be integrated, and four forms are given below:

Equation (3) is used when a fluid is being cooled by a surface of constant temperature, as in a boiler.

Equation (4), when a fluid is being heated up by a surface of constant temperature which, it will be pointed out later, approximately holds in a condenser.

Equation (5), when one fluid is being heated and another cooled by a counter-flow.

The equations are all written in the form for calculating the surface required for a given heating or cooling effect.

$$S = \frac{Wc}{R} \log_e \left(\frac{T_1 - t}{T_2 - t}\right) \quad \text{.....(3)}$$

$$S = \frac{Wc}{R} \log_e \left(\frac{t - T_2}{t - t_1}\right) \quad \text{.....(4)}$$

t = Temperature of saturated steam at the pressure in the condenser.
W = Weight of water per hour.

$$S = \frac{Wc}{R} \left(\frac{T_1 - T_2}{(T_1 - t_1) - (T_2 - t_2)}\right) \log_e \left(\frac{T_1 - t_1}{T_2 - t_2}\right) \quad \text{.....(5)}$$

W = The weight of gas.
w = The weight of water.
c = 0.25 for furnace gases nearly enough.

$$Wc\,(T_1 - T_2) = w\,(t_1 - t_2)$$

provided there is no air infiltration.

Many kinds of formulae of this sort can be obtained by assuming the transfer rate to vary with the temperature in some empirical way, but I do not believe it is ever necessary to use anything other than the equations given, and in most cases the simple Equation (1) is all that is required.

To use these formulae it is necessary to know the transfer rate, and any experimental research should be solely for the purpose of determining this value of R as it varies with different fluids and with temperature. Unfortunately, some experiments have been made from the viewpoint of establishing some general law, and as a result the data obtained is of not much use until it has been recalculated. Even then there is always a question as to whether the temperatures are properly measured, the effect of conduction along the tube eliminated, etc. There are very few accurate determinations. Mostly the variation between experiments supposed to be duplicates of each other are as much as 10 per cent. This is not of much consequence in the practical application of the formula, but it does interfere with any scientific consideration of its variation.

For the purpose of giving values to the transfer rate I wish to refer to three sets of experiments, one set giving the rate for furnace gases in a circular flue made under my direction at Bayonne, the report of which was published in bulletin form by the Babcock & Wilcox Company; second, experiments by T. E. Stanton, published in the Philosophical Transactions of the Royal Society of 1897, giving the transfer rate for water in a tube; and third, to the experiments of Mr. George A. Orrok with the condensation of steam, published in the "Transactions of the American Society of Mechanical Engineers," 1910. I have selected these experiments, as they are, so far as I know, the only ones which cover a sufficient range to be of much value in general consideration of this sort.

The Bayonne experiments were made with a tube two inches in internal diameter. The mean temperature of the gas was calculated from the heat given out, which eliminates the very extensive errors resulting from an attempt to measure this temperature with a pyrometer. If there is any error in the values assigned to the mean gas temperature, it must come from the flow of heat through the tube produced by the internal gas motion that was referred to above. The transfer rates are determined for the full range of temperatures met with in a boiler, and thus constitute the first set of experiments over this range. A full ac-

count of the results and description of the apparatus is given in the report. The results can be summarized as follows:

$$R = 2.1 + 0.000\,626\,[1 + 0.000\,575\,(T - t) - 0.000\,000\,866\,(T - t)^2]w \quad (6)$$

in this equation w is the weight of gas in pounds per square foot of tube sectional area per hour.

It is to be noted that the transfer rate can be expressed by an equation of the exact form that the theory suggests, even though the theory is somewhat questionable. The increase of the constant a with the temperature difference is negligible, but the variation of the constant b is quite marked. The expression given is entirely empirical, and I am quite sure is not as it would be if the exact form were known. Below are two other empirical forms in which the temperatures are in degrees Fahrenheit (absolute):

$$R = 2.1 + \frac{w}{1,650}\,\frac{T}{T - t}\log_e\left(\frac{T}{t}\right)$$

$$R = 2.1 + 0.00193\left(\frac{T + 360}{T + 2,500}\right)w.$$

In the first of these expressions the constant b is of the form suggested for the complete transfer rate coefficient by Professor Fessenden in a paper read before the "American Society of Mechanical Engineers" last Spring. This is a one-constant equation, and ordinarily would be considered as of more importance than any other form of expression. I believe, however, that it is simply coincidence, as I cannot conceive of the effect of the variation with temperature, density, conductivity and viscosity being grouped together and controlled by one constant. Since further there is no theoretical reason for assuming this form, and as it is impossible to integrate the general equation with its use, I give it here, together with the other, simply to emphasize how easy it is to get formulae for temperature variations and for heat absorption that differ from the simpler equation given, only in the assumption made as to the form of the expression for the heat transfer rate.

The Bayonne experiments do not show the variation with tube temperature. It was originally intended to obtain this data, but the tube temperature had to be kept around 180 degrees F. to prevent trouble from moisture in the gases, and this did not leave enough range with the way the apparatus was designed to make it worth while attempting a second set with different tube temperature. For the investigation to be complete this should be determined. For all ordinary purposes, however, I am inclined to believe the first value for the transfer rate can be used even with different tube temperatures without serious error. It probably also can be used with tubes of different diameter.

When the gas flow is over tubes at right angles to their length, as with horizontal boilers cross baffled, the mean transfer rate is very nearly

$$R = 2.0 + 0.0014\,w \quad (7)$$

In this expression w is equal to the total weight of gas divided by the net area between the tubes in one row as averaged between the different passes. The value of b is more than the value of the constant in the expression applicable to a circular flue, and is due probably to the effect of staggering the tubes, but it is to be remarked that for the same velocity the friction loss would be more, so that for the same friction loss the value may not be much different.

At very high velocities the transfer rate is almost directly proportional to the gas flow. When this is true and the corresponding value of R is

substituted in Equation (3) the gas weight cancels out, and under these conditions a given area of surface would cool a hot gas the same amount, regardless of the amount of gas that is swept over the surface. At lower velocities the first term in the expression for the transfer rate is relatively of more importance. In boiler work around rating the gas flow is usually about 1,000 pounds per square foot of net area per hour. The corresponding transfer rate is 3.4, of which amount 2.0 is from the constant term and 1.4 from the term that varies with the gas flow. It is, therefore, to be expected that flue gas temperatures should increase as the capacity increases in all cases, and this is, of course, true. The second term of this heat transfer equation is the one which varies with temperature, so that, when it is small, the variation with temperature is relatively small, and Equation (3) can be used in plotting temperatures through a tube in a flue boiler or between the tubes in a water tube boiler. This temperature curve is only of academic interest. Equation (3), together with the equation for the heat transfer rate and the equation for the heat transmitted by radiation from the furnace, gives all the formulae necessary for design work in either type of boiler. There is no such a thing as a theoretical length of tube or of gas travel for the best results. The greater the heating surface and the longer the gas travel, the lower the flue gas temperatures will be and the higher the efficiency. Against this must be balanced the increased first cost and the greater draft required, and the arrangement finally selected must be made from a commercial and operating viewpoint.

If the value of the transfer rate is much affected by the tube temperature the transfer in economizers should be somewhat lower than it is in boilers, and ordinarily transfer rates of only one-half to two-thirds of the amount are used. I believe, however, that the same transfer rate could be used without much error when the tubes of the economizers are staggered and air leakage is entirely excluded. I have had an opportunity recently of testing a Green economizer of this sort, and the heat absorbed was very nearly as much as is absorbed by boiler surface with the same gas flow. In economizer work the approximate formula given in Equation (1) can usually be used without much error. If the flow is counter current, truly speaking, the exact formula to use would be Equation (5). The flow, however, is never just like this.

If, in place of the transfer rate R its reciprocal is used, an expression is obtained which measures the resistance to heat transmission which is directly analogous to the resistance in electrical work, the difference in temperature corresponds with the difference in voltage and the heat flow to the electric current. The total resistance in a boiler or economizer is the sum of three separate and distinct terms:

First. The resistance to the passage of the heat from the gas to the tube surface, which is measured by the reciprocal of the value of R just discussed.

Second. The resistance in the tube itself, which is equal to the thickness of the tube in inches divided by the coefficient of conductivity as discussed in the first part of this paper.

Third. The resistance from the tube to the water.

The tube resistance is negligible in comparison with the resistance on the gas side, and it will be seen later on that this also is true for the resistance on the water side so that the total resistance is very nearly equal to the resistance on the gas side. This warrants the use of formulae without reference to the velocity of circulation of the water.

The experiments of Stanton on the rate at which water will absorb heat from a tube were made on three sizes of tubes and approximately, at least, the temperature of the tube surface was observed. His experimental results did not always check one with another and there is some question as to just the amount of surface engaged in the transfer.

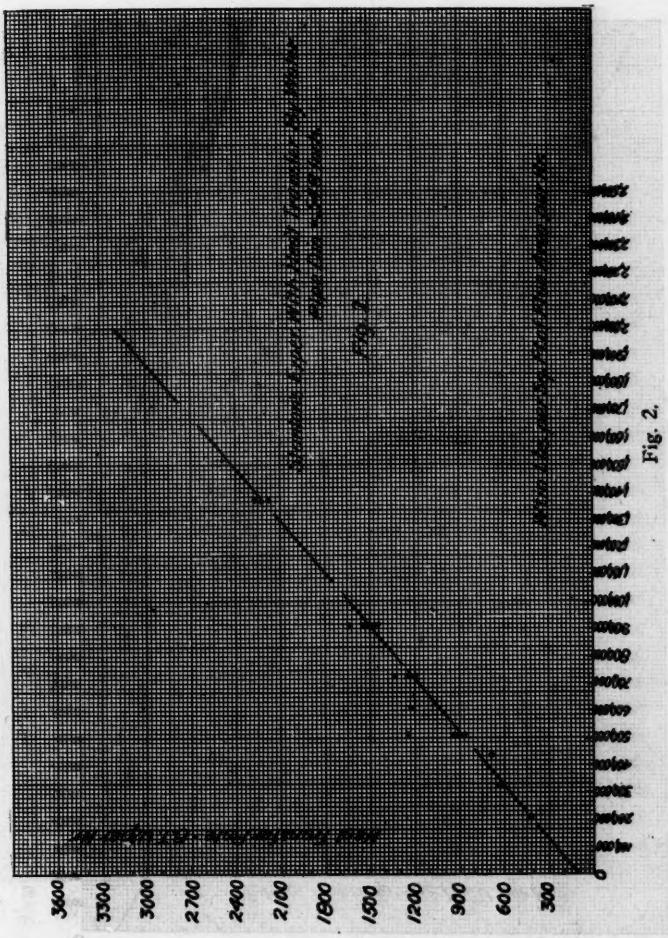

Fig. 2.

There was also some transfer of heat at the ends due to conduction. The results were worked up to prove a formula suggested by Professor Reynolds from the formula he had given for the friction loss in pipe lines, which is now known to be incorrect, so that Stanton's paper is of little value with the results as given. I have taken his data, however, and calculated heat transfer rates, which show that very accurately they are a linear function of the mass flow, and can be noted in Figs. 2, 3 and 4. There is some variation of temperature, but the temperature range was not wide and this variation is not excessive. A more marked variation is with the size of the tube. Stanton, in his paper, seems to believe that the tube size made little difference, but when the results are

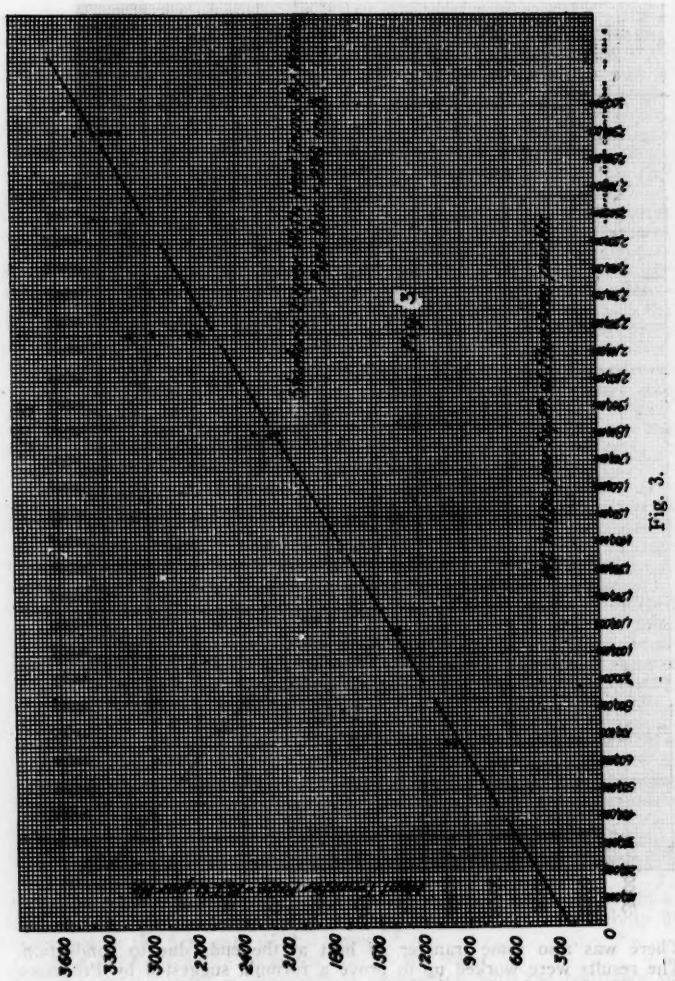

Fig. 3.

refined as they are in calculating transfer rates this is shown not to be true. Disregarding the temperature variation, Stanton's experiments make

$$R = 110 + 0.00208 \left(1 - \frac{0.139}{d}\right) w \quad \ldots\ldots\ldots\ldots\ldots\ldots (8)$$

The constant b is about three times what it would be for air in the same tubes, and this is not widely different from the ratio of the specific

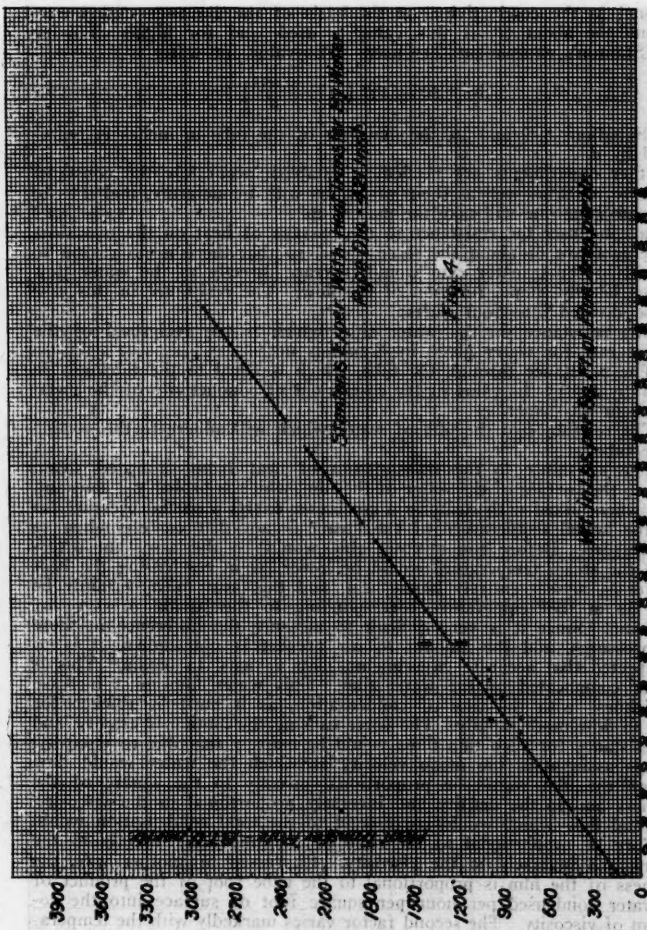

Fig. 4.

heats. The mass flow, with a fluid as dense as water, runs into large numbers and, since the density does not vary, it is more convenient to use v, the velocity in feet per second, in place of w in the value for R, which then becomes

$$R = 110 + 465 \left(1 - \frac{0.139}{d}\right) v \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

If the errors in Stanton's experiments are what I conceive them to be, the values of the transfer rates at low velocities are too high, which would mean that the constant a is too large. If this constant is pro-

portional to the conductivity, on the basis of the Bayonne experiment, it would be about 50, if this value is used, and for higher transfer rates Stanton's experiments are used. The value for water is

$$R = 50 + 467\left(1 - \frac{0.133}{d}\right)v \qquad (10)$$

In Equations 8, 9 and 10, the diameter of the tube in inches is represented by d.

Further experiments are needed to establish accurately the constants in this formula, as well as their variations with temperature, and it is to be hoped some one will take the matter up.

In a condenser the resistance to the passage of heat on the two sides of the tube are of about the same order, so that there is no reason to expect that the transfer rate would increase directly as the water flow or even be any simple function of the flow. The resistance in this case can be divided into four parts:

1. The resistance on the water side, which is the reciprocal of the transfer rate for water given in Equation 10.

2. The resistance through the tube, which is equal to the thickness in inches divided by the value of the co-efficient of conductivity of the metal as used in this paper.

3. The resistance of the film of water which will adhere to the outside of the tube.

4. The resistance to the transmission of heat from saturated steam vapor to the surface of the water of the film, which is probably a constant.

Air when mixed with steam in a condenser exerts a partial pressure, so that the actual temperature of the steam is not that corresponding to the pressure. Before the mixture this would be the case, but afterwards the steam temperature would be lower, as in condensing the heat transfer is almost entirely latent heat, and this heat must certainly be given up at the temperature corresponding to the partial pressure of the steam. Since it is very difficult to measure the difference in these two pressures, and the amount of air present is not known, it is to be expected that accurate experimental data is hard to obtain. This is further complicated by the water film. This water film must increase in thickness with the rate of condensation, as it must be thicker for water to flow more rapidly. The force causing the motion is gravity, and the resistance for such thin films would be entirely that due to viscosity. On this assumption the mean velocity at which the water would flow through the film would be proportional directly to the square of the thickness and to the density and inversely proportional to the viscosity. The total flow which would correspond to the total condensation would therefore be proportional to the cube of the thickness, directly to the density, and inversely to the viscosity. Expressed in another way, the thickness of the film is proportional to the cube root of the product of the water condensed per hour per square foot of surface into the co-efficient of viscosity. The second factor varies markedly with the temperature, and this must be taken into account. The viscosity of the fluid next to the surface is of the most importance, and the temperature used in determining it should be that of the external surface of the tube.

Using letters with the following meaning the transfer rate for condenser tubes can be written down:

r_w = the transfer rate from the water to the tube, which can be taken as equal to the value R given by Stanton's experiments.
r_s = the transfer rate from the steam to the water film, which is taken as constant.
d_1 = the internal diameter of the tube.
d_2 = the external diamter of the tube.
k = the conductivity of the metal of the tube.

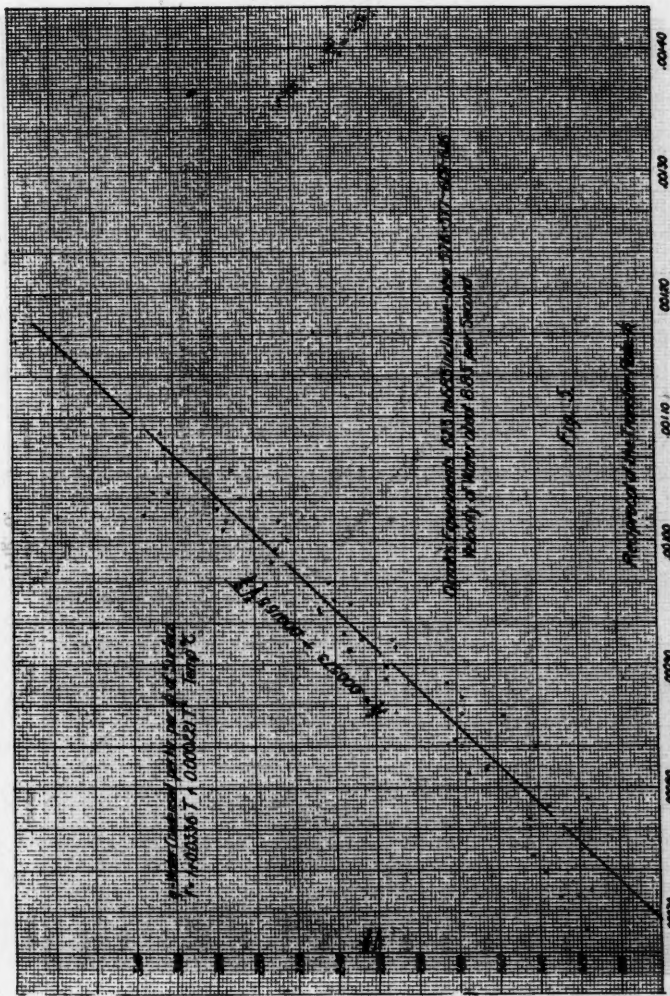

Fig. 5.

t = the tube thickness in inches.
q = the weight of steam condensed per hour per square foot of surface.
f = reciprocal of the temperature factor of the coefficient of viscosity of water.
j = a numerical constant.

$$\frac{1}{R} = \frac{1}{r_w} + \frac{d_1}{d_2 r_2} + kt + j\sqrt[3]{\frac{q}{f}} \qquad (11)$$

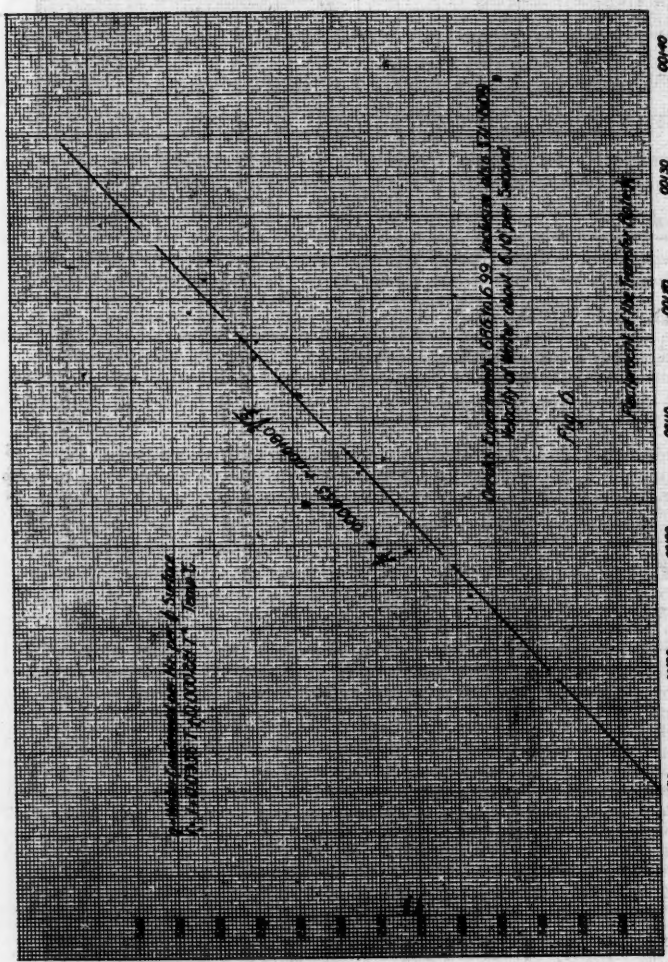

Fig. 6.

It will be noted that with a constant water velocity the reciprocal of the total transfer rate should be a linear function of the cube root of q divided by f. Mr. Orrok's experiments have been referred to to check this up, and in Figs. 5, 6 and 7 are plotted values taken from the report of his experiments. The value of f used was

$$1 + 0.0336\,T + 0.00022\,T^2 \quad \ldots\ldots (12)$$

in which T is expressed in degrees Centigrade.

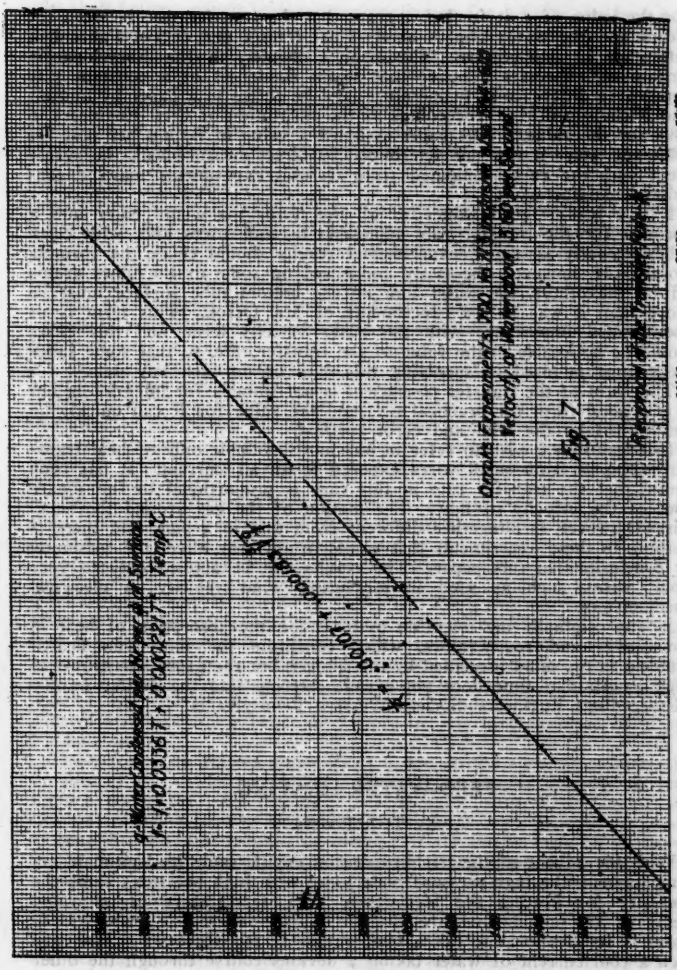

Fig. 7.

Only the last series of tests was used, as there seems to be some errors in the earlier tests. It is evident that the law assumed holds closely for the velocities considered; at lower velocities the character of the motion of the water in the tubes changes as an approach is made to the critical value. When it is further considered that any variation in the conductivity of the water with the temperature would throw this out, it appears that the conductivity must be nearly constant. The values obtained from this plot gives j the average value 0.00017.

Taking the conductivity of the tube at 0.0005 and using Equation (10)

for the value of r, it further appears that the transfer rate between the steam and the water film is about 3,400 B.t.u.'s per degree Fahrenheit difference in temperature. Professor Callender, in the Encyclopaedia Britannica, expresses a belief that steam condenses at a constant rate per degree difference in temperature, and he refers to a paper by himself and Nicholson in the Proceedings of the Institution of Civil Engineers in 1898 for the value.* This paper assigned a value in B.t.u.'s per hour of 2,664 and, since it was not determined by direct measurement, the agreement with the value computed from Orrok's experiments may be considered satisfactory.

I have just assigned values for all the terms that would enter into a formula for the transfer rate applicable to condensers and, with clean tubes properly arranged, experimental results should check the formula closely. If, however, the steam condensed on some of the tubes drains off on the others, the water film on the bottom tube surface must be more than it was in the experiments of Mr. Orrok. This term is of about equal importance with the water velocity term, and this effect would be noticeable in the value of the transfer rate, so that some judgment must be used in supplying this data or, in fact, any data.

The form of the transfer co-efficient is not suitable for integration, but it is a good form for practical use, and the value for R can quickly be determined by the trial and error method.

In conclusion I wish to refer to one more example only of heat transfer. This is from small wires exposed to an air current. It has been found that the transfer rate in this case is proportional to the square root of the velocity of flow. The motion of the gas around the small wires, however, is altogether different from that in a pipe, and there is nothing in these experiments that would in any way affect the reasoning or conclusions in this paper. The results can be applied only to very small wires, such as used in measuring wind velocity in electrical anemometers.

It was originally my intention to touch on the absorption of heat in hot stoves and regenerators, which constitute the most complicated problem in my mind presented by the subject, but the paper has grown to such length that time does not permit entering into the matter now.—"Proceedings Engineers' Society Western Pennsylvania."

FEED-WATER HEATING.

By T. J. Rogers.

With the advent of the triple-expansion engine came its legitimate accompaniment, the high-pressure boiler, and with high pressures came the necessity for heating the feed-water. The temperature of saturated steam of 175 pounds pressure absolute is 370.8 degrees F., while feed-water is frequently at so low a temperature as 110 degrees F. We have then a cool current of water taking a devious course through the other water of the boiler, impinging in its wanderings against hot heating surfaces, subjectng them to sudden contraction or against columns of steam bubbles on their way to the water line, effecting their condensation, and perhaps inducing foaming. In fact, a current of cold feed-water must be rated as one of the worst disturbing elements that a boiler has to contend with.

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* "Proceedings, Institution of Civil Engineers" (British), "On the Law of Condensation of Steam Deduced from Measurements of Temperature—Cycles of the Walls and Steam in the Cylinder of a Steam Engine," Vol. CXXXI, pp. 147, 206, 234, 260.

In shell boilers, which have neither appliances for heating the feed nor means for producing a thorough circulation, a body of comparatively cold water settles at the bottom and does considerable mischief. The general double-ended boiler attains a length of about 20 feet. In the case of such a boiler, if the temperature of the water at the bottom be 150 degrees and the temperature of the upper portion of the shell be 370 degrees F. (corresponding to steam of 160-pound gauge pressure), the difference of expansion between the two opposite parts of the shell amounts to about 0.33 inch. The strains thus set up are well known to be the cause of many leaks and fractures.

With feed-water heated to live steam temperature, such strains are entirely obviated. In some cases the feed-water is not heated up to the live steam temperature, the benefits of such treatment being, of course, proportionately less.

For heating feed-water three methods have been tried—namely:

Class A. Arrangements of pipes or flues in uptake.

Class B. Arrangements of feed coils surrounded by exhaust steam from receivers, or live steam.

Class C. Direct commingling of the steam with the feed-water without the intervention of heating surfaces.

Devices in Class A have long been in use. The chief objections to them are: Liability of the heating surfaces to burn from being fouled with scale, grease, or other dirt; liability to corrosion from the effects of moisture, acids in the furnace gases, and so on; and also inaccessibility for effecting repairs.

The other two forms—Classes B and C—are the ones now most commonly in use, and it is not improbable that uptake heaters will be entirely discarded, if, indeed, such is not the case at the present time.

For Class B it may be said that the arrangement of tubes acts to a certain extent as a grease extractor, the grease adhering to the tubes, from which it may be moved periodically. With the increasing employment of oil eliminators on the exhaust pipes of engines, it is not likely that the feed-water will contain much, if any, oil. Should the feed-water contain any scale-making constituents, the hot tubes will precipitate and receive the same, and thus save the heating surfaces of the boiler. Against this class of heater may be noted its somewhat complicated structure, its expense, and more or less inconvenience of cleaning and scaling.

Those forms of heaters under Class C are, in the author's opinion, the simplest of all, just as the old jet condenser was simpler than the modern surface condenser. By heating the feed-water with direct contact of steam a smaller apparatus is made effective, since there is no transmission of heat through slow conducting metals. If the heater is to be employed in the double capacity of heater and scale precipitator, then there must usually be provided a series of plates or pans to receive the precipitate, the heater being opened at intervals and the pans removed for scaling. In this case there is probably but little saving in weight over Class B. The scale pans of Class C are somewhat more convenient for removal and scaling than are the tubes of Class B, and the matter of expense is probably in favor of Class C.

Since steam is used as the heating agent in two of the classes of heaters under consideration, it becomes important to inquire whether the heating should be accomplished entirely with live steam or partly with exhaust steam and partly with live steam. We shall first assume that 1 pound of feed-water is heated up to the live-steam temperature by the employment of live steam alone. Next we shall consider 1 pound of feed-water heated up in successive stages by steam from the low-pressure receiver, steam from the intermediate-pressure receiver, and finally by live steam.

1. To heat 1 pound feed from 110 degrees F. to 370.8 degrees F., with steam of 175 pounds absolute pressure and temperature of 370.8 degrees F.:

Number of heat units to be imparted = 260.8 degrees F.
Number of heat units to be surrendered = 855.6 x = 260.8 degrees F.
Number of pounds steam condensed = x = 0.3048.
Total pounds feed sent to boilers = 1.3048.

2. (*a*) To heat 1-pound feed from 110 degrees F. to 228 degrees F., with steam from the low-pressure receiver, of 20-pound absolute pressure and temperature of 228 degrees F.:

Number of heat units to be imparted = 118.
Number of heat units to be surrendered = 954.9 x = 118.
Number of pounds of steam condensed = x = 0.1235.
Number of pounds of augmented feed-water = 1.1235.

(*b*) To heat 1.1235-pound of feed from 280 degrees F. to 302.9 degrees F., with steam from the intermediate-pressure receiver of 70-pound pressure absolute and temperature of 302.9 degrees F.:

Number of heat units to be imparted = 84.15.
Number of heat units to be surrendered = 902.9 x = 84.15.
Number of pounds of steam condensed = x = 0.0932.
Pounds of augmented feed-water = 1.1235 + 0.0932 = 1.2167.

(*c*) To heat 1.2167-pound of feed from 302.9 degrees F. to 370.8 degrees F. with steam from boiler of 175-pound pressure absolute and temperature of 370.8 degrees F.:

Number of heat units to be imparted = 82.61.
Number of heat units to be surrendered = 855.6 x = 82.61.
Number of pounds of steam condensed = x = 0.0965.
Total feed = 1.2167 + 0.0965 = 1.3132 pounds.

In Case 1, 1 pound of steam passed successively through the high-pressure, the intermediate-pressure, and the low-pressure cylinders. In Case 2, 1.2167 pound of steam passed through the high-pressure cylinder, 1,1235 pound of steam passed through the intermediate-pressure cylinder, and 1 pound of steam passed through the low-pressure cylinder. Since, in all properly adjusted triple-expansion engines, the work is divided more or less equally among the three cylinders, we may, for the purposes of comparison, assume that 1 pound of steam performs the same amount of useful work in each one of the cylinders through which it may pass. Then, in case No. 1, the work would be represented by 1 + 1 + 1 = 3. In case No. 2 the work done would be represented by 1.2167 + 1.1235 + 1 = 3.3402.

Placing the above results in tabular form we have—

	Heating by Live Steam Only.		Progressive Heating.
Total pounds feed sent to boiler.....	1.3048	..	1.3132
Number heat units surrendered to heat feed..	260.8	..	284.76
Number pounds steam condensed to heat feed	0.3048	..	0.3132
Comparative work done in cylinders.....	3.00	..	3.3402
Work done per pound of feed.....	2.299	..	2.543

Gain by progressive heating, 10.6%.

Whether the increased economy to be anticipated by the adoption of progressive heating would warrant its increased cost is an open question. As to the absolute economy of feed-water heating, it is not difficult to understand why the uptake heaters effected a saving, as the heat which they utilized would otherwise have passed up the smoke pipe and been wasted. Steam feed heaters were at first employed to relieve the boiler strains consequent upon the employment of cold feed-water, and the re-

sultant economy of the practice was a surprising discovery.—"International Marine Engineering."

BEARING LUBRICATION.

INFLUENCE OF SURFACE VELOCITY ON MEAN FILM THICKNESS.

By Boynton M. Green,* Pittsburgh, Pa.

Assuming that a machine will do the work for which it is designed, its economic value depends on the degree of efficiency with which it will do that work. The energy supplied to it is consumed in three ways: in overcoming the external load, or in doing the actual work required of the machine; in deforming the various parts of the machine, and in overcoming friction. The energy consumed in doing the actual work required is a legitimate expenditure for which there is value received. If the machine is properly designed, the energy lost in deforming its parts is returned somewhere during the work cycle, because the parts will never have been stressed beyond their elastic limit and hence, on returning to their original shape, will liberate the energy spent in deforming them, while that spent in accelerating masses will be given back upon their return to their original velocities. This leaves the energy lost in friction, which is a direct waste. Some friction is inevitable, so the question is: how may this loss be reduced to a minimum?

HISTORICAL DEVELOPMENT.

It is interesting to note that this important problem of design was the last one to be attacked from a scientific standpoint. For years after the designer was able to calculate with a fair degree of accuracy the stresses in frames, shafts, levers and gears, he was content to design his bearings by guess and precedent. This tendency to slight bearing design was fostered by low machine speeds and lack of efficiency data. But with the introduction of electricity as a motive power and the resulting higher speeds and ever-increasing demand for greater efficiency, the necessity for better bearing design became urgent.

The first important attempt to investigate the conditions obtaining in a bearing was made in 1883 by Beauchamp Tower, an Englishman, at the request of the Institution of Mechanical Engineers. He used a journal 4 inches in diameter by 6 inches long, the bearing only covering the upper half of the journal. He was able first to explain the difference between partial or greasy lubrication and complete or flooded lubrication. Tower's results were qualitative rather than quantitative, but they brought out the facts that there was probably a complete film of oil separating journal and bearing in the case of flooded lubrication, and that the conditions of bearing friction in the case of flooded lubrication approximate fluid friction more nearly than solid friction.

The next step was made in 1885 by Prof. Osborne Reynolds, who used Tower's numerical data as the basis of a mathematical discussion of the subject. Reynolds applied a hydrodynamic theory to Tower's data and obtained an equation between the variation of pressure over the surface and journal velocity, which explained the existence of the oil film at a high pressure. He showed the presence of a wedging action of the lubricant, and this in turn brought out the importance of the bearing allowance, or difference in diameters of journal and bearing. From this followed the discovery of the general law for pressure distribution throughout the oil film, and the fact that the point of nearest approach

*Leland Stanford University.

of journal and bearing changes position with change of load. Reynolds realized that viscosity changes with temperature, so he made a determination of the relation between viscosity and temperature for olive oil, and deduced an empirical formula from which he obtained expressions for the approximate variation of viscosity with speed and load, since both these affect the bearing temperature. These expressions brought Tower's results into very close agreement with Reynolds' hydrodynamic theory. The hydrodynamic theory of fluid friction was also developed independently in 1884 by Petroff, a Russian, (data published in German in 1887).

This theory is undoubtedly the correct one to apply to the case of flooded lubrication. However, it cannot be applied directly, as the equations contain several constants which can only be determined by experiment, and it is to this work that later investigators have turned their attention rather than to the development of new theories. In 1903 some important investigations were carried on by O. Lasche for the Allgemeine Electricitäts-Gesellschaft of Berlin. Up to that time investigations had only been made with loads under 500 pounds per square inch and velocities under 500 revolutions per minute (among the most important were those by Stribeck, Z.d.V.d.I, 1902), and the A.E.G. found it necessary to obtain data for velocities up to 3,000 revolutions per minute and correspondingly high loads. Lasche's work was quite exhaustive and incidentally threw considerable light on the transmission of heat away from the oil film.

FLUID FRICTION.

The general equation for fluid friction is

$$\mu = \eta \, V/py$$

where μ = coefficient of friction
η = coefficient of viscosity
V = journal surface speed
p = pressure per unit of projected area of bearing
y = mean film thickness.

Usually p and V can be determined from the conditions of the problem and values of η can be taken from known data on oils. Concerning values of y, nothing definite is known, but some general deductions can be made. According to Smith and Marx's Machine Design, 1915, it is a function of the running-fit allowance, of p, of V, and of the temperature t of the bearing, which may be summed up in the following expression

$$y \frac{k_a \, V^x}{p^w \, t^z}$$

From a consideration of this expression and its application in the general equation for friction it would seem that the next step would be an investigation of the mean thickness of oil film Some work was done on this subject in 1897 by Professor Kingsbury (Jour. Am. Soc. Nav. Eng.), but as he used air as a lubricant, his results can only be taken as an indication of what to expect when using oil. As the mean film thickness y is influenced by four variables, it would be necessary to make the investigation in four steps. In Kingsbury's experiment the velocity only was changed. The pressure and allowance were kept constant while the apparatus was allowed to run until the temperature became constant before any test readings were taken.

EXPERIMENTAL PART.

In the present experiment it was thought best to follow Kingsbury's lead and investigate the influence of velocity on mean film thickness, and the apparatus was designed with this point in view. The bearing is shown in section in Fig. 1; it is non-adjustable and consists of a plain phosphor-bronze sleeve about 3¼ inches in diameter by 7 inches long. pressed into a cast-iron housing which was bolted to a lathe bed. Lubrication was effected by two steel oil rings of rectangular section. Two sets of oil grooves were cut in the upper half of the bronze as shown in the figure. On assembling the bronze sleeve and housing, it was found that the bronze was elliptical in section, with the major axis vertical, the average difference in diameters being 0.001 inch. To obtain the best results from the apparatus, the sleeve should have been reamed after pressing it into the housing. Although this was not done, the inaccuracy did not seem to affect the results appreciably.

The journal was a piece of mild steel, ground carefully to a diameter of 3.244 inches, giving a running-fit allowance of 0.0035 inch, or about 0.001 inch per inch of diameter. On each end of the journal was fitted a cast-iron flywheel weighing about 67 pounds, secured by a nut. The total weight of the assembled journal, flywheels and nuts was 165.5 pounds, making a nominal load on the bearing of 7.275 pounds per square inch of projected area. The whole apparatus was mounted on two parallel lathe beds, and three lathe heads with four-step pulleys were utilized for the drive. The usual overhead countershaft drove the first lathe head, which drove the second head by a short piece of shaft and dogs. The second head drove the third by a short belt and tightener pulley. It was found necessary after several preliminary trials to introduce the tightener, so that the second belt could be used for gradual acceleration of the heavy rotating mass. The third head drove the test journal by a piece of ¼-inch steel shaft about 2 feet long. As this flexible drive rod was loosely connected by a dog to the lathe head and by a cotter pin to the journal, it is fairly certain that no external deflecting load was applied to the latter. With the drive arranged in this way, sixteen speeds were possible, but as all the cone pulleys were of the same design, some of the speeds were duplicated, so that only six speeds could be used.

With this apparatus it would be possible to vary the bore allowance by starting with a very small allowance and then grinding down the journal to give larger allowances. This procedure was contemplated when the apparatus was designed, but lack of time prevented tests with more than one allowance. To locate the position of the journal with relation to the bearing, three micrometers of a design similar to Kingsbury's were used. The micrometers and journal were placed in the primary circuit of an induction coil with a battery, and a telephone receiver was connected to the secondary of the coil. With Kingsbury's apparatus it was only necessary to mount one micrometer in the bearing, because his bearing was a simple cast-iron cylinder, so supported that it could be rotated about its own axis, thus bringing the micrometer into any position desired. His method was to locate the point of nearest approach and read the micrometer, calling this the zero reading. Then the bearing was rotated 180 degrees and another reading was taken. Half the difference between the zero reading and the second reading gave the radial distance between the axis of the bearing and the axis of the journal, and the angular displacement was read from graduations on the end of the bearing cylinder.

In the present experiment the bearing could not be rotated, so it was necessary to use three micrometers spaced 120 degrees apart around the bearing in a plane through its center and perpendicular to its axis. The moving part of the micrometer consisted of a ⅛-inch steel rod hardened

at the contact end and threaded 40 threads per inch at the other end. To the threaded end was clamped a pointer 3¼ inches long which served both to turn the rod and to indicate the reading. The rod was carried in a brass sleeve threaded to receive it and electrically insulated from the bearing by a fiber bushing pressed on to the sleeve. The graduated scale was carried on an arm clamped to the top of the brass sleeve. This arm also carried a binding post for the electrical connection. The whole micrometer was held in place in the bearing by means of a brass collar which screwed down over the tapered fiber bushing.

The micrometers were calibrated by screwing them into a ring and then screwing down the micrometer point to touch a standard inside micrometer. The experimental micrometers were graduated in this way to thousandths and the ten-thousandth divisions were laid off from these

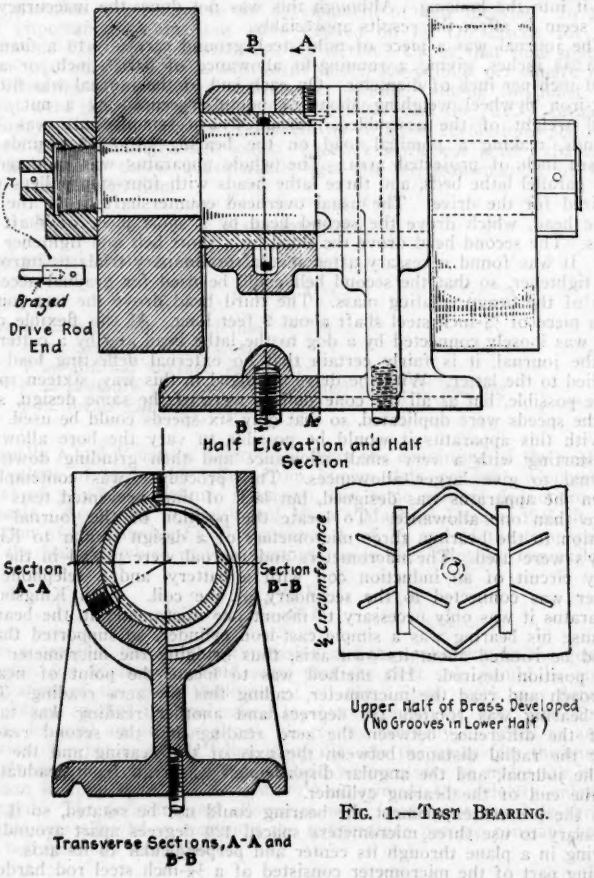

FIG. 1.—TEST BEARING.

major divisions. Micrometers 1 and 3 were graduated for a total of three thousandths, and micrometer 2 was graduated over a range of six thousandths. Before any test readings could be taken, it was necessary to set the micrometers to zero, *i. e.*, bring the micrometer points flush with the inside surface of the bearing brass by placing inside the bearing a sector cut from a polished iron ring which had been ground to the exact diameter of the bearing. The zero readings of all the micrometers were checked after every three or four test readings.

TABLE 1 DISTANCE BETWEEN AXES

Average Experimental Data					Results	
V Ft. per Min.	Temp. deg. fahr.	Micrometer Readings, In.			Axial Dist. In.	Angle
		1	2	3		
0		0.0002	0.0035	0.0002	0.00175	90°00′
196	58	0.0001	0.0026	0.0006	0.00158	80°45′
306	66	0.0003	0.0025	0.0004	0.00153	88°10′
511	66	0.0003	0.0025	0.00065	0.00142	82°00′
817	78	0.0002	0.0023	0.0005	0.00134	83°00′
1224	70	0.00025	0.0021	0.0004	0.00125	86°05′
2042	80	0.0002	0.0020	0.0006	0.00113	77°40′

TABLE 2 MEAN THICKNESS OF OIL FILM

V	Distance between Axes	Min. Thickness Film	Mean Thickness Film
0	0.00175	0	0.000875
196	0.00158	0.00017	0.000960
306	0.00153	0.00022	0.000985
511	0.00142	0.00033	0.001040
817	0.00134	0.00041	0.001080
1224	0.00125	0.00050	0.001125
2042	0.00113	0.00062	0.001185

The runs were made short purposely, usually under two minutes, to eliminate the temperature factor as far as possible. A long run was made at a constant speed to obtain some information concerning the influence of temperature on the position of the journal relative to the bearing, but the micrometer readings showed that the bearing itself was expanding very rapidly due to its thin section, so this investigation had to be abandoned. From ten to fifteen readings of each of the micrometers were taken at each speed and the most consistent of the readings at each speed were averaged to obtain the final experimental data which are given in Table 1.

In determining the mean thickness of oil film from the results the following approximations were made: (*a*) that the loaded portion of the film was that below a horizontal plane through the center of the bearing, (*b*) that the thickness of the film at this plane and on each side of the journal was equal to the radial bearing allowance, (*c*) that the mean

thickness of film was the average of the thickness at this plane and the minimum thickness of the film. The minimum thickness of film is the radial allowance minus the distance between the axes of the journal and bearing. Making use of these approximations the resulting values of mean thickness of film are given in Table 2.

The general equation of the curve plotted is

$$y = b + c^a \sqrt{V} \quad \ldots\ldots(1)$$

where y = mean thickness of film in inch
b = one half the radial allowance
c = constant dependent on allowance and possibly on viscosity
a = constant dependent on viscosity and possibly on allowance
V = surface velocity of journal in foot per minute.

By translating the horizontal axis of the curve the constant b is eliminated and the equation becomes

$$y = c^a \sqrt{V} \quad \ldots\ldots(2)$$

From the experimental curve the values of c and a were found to be

$$c = 0.0000049, \; a = 1.8$$

and the resulting empirical equation is

$$y = 0.0000049^{1.8} \sqrt{V} \quad \ldots\ldots(3)$$

As to the application of the empirical equation in the general equation for bearing lubrication, it may be said that it may be used directly for conditions of the same allowance and a lubricant having the same viscosity. For other allowances and lubricants the change of c and a cannot be stated definitely. It can be said in general that c will change with some function of the allowance and a with some function of the viscosity. It is also possible that c may be affected by the viscosity and a by the allowance.

The oil used during the experiment was an ordinary mineral oil, known as Vacoline, manufactured by the Standard Oil Company. Its viscosity as given by the Engler viscosimeter was 11.0 at 20 degree Centigrade and 2.95 at 50 degrees Centigrade compared with water at 20 degrees Centigrade, which gave a specific viscosity of 38.9 at 20 degrees Centigrade and 8.7 at 50 degrees Centigrade.—"Journal Am. Soc. M. E."

SUGGESTIONS FOR SELECTING AND TESTING DRILL STEEL.

By Frank H. Kaiser.*

Mining with rock drills has progressed to a point where the actual efficiency of the drilling operation depends more on the drill steel than ever before. In order that the steel can meet these conditions, one must be able to obtain a uniform quality of steel and be able to subject the steel to the proper treatment. There must be means of doing this from day to day. As a suggestion for meeting these conditions the facts in this article are presented.

Although the chemical qualities of steel have a great influence, the analyses of drill steel are practically standardized. A chemical laboratory would be a good asset, but it is not essential for this work. When the proper heat-treatment has been given to the steel—assuming the steel to have the chemical properties as specified for standard drill steel—and the

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treated steel is either too soft or gives exceptional service, a sample of it could be sent to a commercial laboratory for chemical analysis.

Drill steels are made by both the openhearth and crucible methods. The crucible-made are lower in phosphorus and sulphur as a rule, while the openhearth is lower in silicon. For each grade in either make, the carbon content will vary according to the size of the steel. In general the average analysis of a drill steel would show:

	%		%	
Manganese	0.25	to 0.40.	Some makes as low	as 0.15
Phosphorus	0.01	to 0.025.	Some makes as high	as 0.04
Sulphur	0.015	to 0.025.	Some makes as high	as 0.04
Silicon	0.10	to 0.20.	Some makes lower	
Hardness (as received)	40 to 50 scleroscope or 225 to 300 Brinell			

The carbon content will vary as follows, as to shape, size and make:

	% Carbon
American-made octagon	0.80 to 1.00
American-made cruciform	0.80 to 0.95
American-made hexagon and round hollow	0.70 to 0.85
American-made solid and hollow spiral	0.70 to 0.85
F.J.A.B. No. 7	0.65 to 0.80
F.J.A.B. No. 10	0.90 to 1.05

It is advisable in selecting a drill steel that either a steel company or a drill manufacturer be consulted, as their experience fits them to suggest the best steel for the conditions. Request the steel company to give the critical range of the steel and the proper forging, annealing and hardening temperatures.

Each bar of drill steel as received should have a 3- or 4-inch test piece cut off and the piece and bar numbered. In this way a record can be kept of when the bar was received, from whom and the service given. Determine the Brinnell hardness on the cut end of the test piece and then subject the piece to a simple forging operation, heating to the recommended forging temperature. Anneal the piece as recommended and then harden, using the recommended temperature and quenching medium. If the steels as used are drawn or tempered, the test piece should be heated to the same drawing temperature. Determine the Brinell hardness on this properly treated test piece, then break it and examine the fracture, noting the grain. If the fracture shows a seam or pipe, the bar should be rejected.

Each drill steel made should bear the same number as the bar from which it is made, and a record should be kept of the service of the drill steel. In this way a record will be had of the Brinell hardness of the steel as received, the complete heat-treatment, the hardness of the heat-treated steel and the service of the steel in the mine. After a hundred or so tests have been made, some of the high, medium and low Brinell test bars and those giving the best service can be analyzed. In this way a standard specification can be determined for the drill steel best suited to meet the conditions of the mine, giving the desired carbon range (10 points) and the Brinell hardness range.

The essential equipment consists of the following: Brinell testing machine; furnace for heating the steel, equipped with pyrometer; quenching tank; tempering or drawing bath; means of forging. Except the testing machine, this equipment can be found in the blacksmith shop.

HINTS AS TO APPARATUS.

Gas or oil-fired furnaces of the muffle or semi-muffle type give the best results and are easily manipulated so that a constant temperature can be maintained. Furnaces should be adapted to meet the conditions.

Pyrometers are essential. Base-metal couples should be used either with galvanometer or potentiometer. Use equipments that automatically take care of the cold junction. Both the Wilson-Maeulen base-metal equipment and the Leeds & Northrup potentiometer equipment have proved serviceable. Adapt the pyrometer equipment to the conditions.

Place the base-metal couple in an X or XX iron pipe plugged or welded shut at one end, then place this in the chamber of the furnace as near to the drill steel as convenient. The indicating instrument should be somewhat away from the furnace, so placed as to be free from vibration and excessive changes in temperature.

The pyrometers should be checked weekly at least. One base-metal couple can be kept as a standard and compared with the one in service. When the regular one shows too great a variable error, use the standard as regular and purchase a new couple to use as standard. The indicating instrument should be sent away periodically to be tested and regulated. Complete information and instructions should be obtained from the manufacturers at the time of purchase.

It is not essential to have an equipment for determining the critical range. Obtain this range from the steel maker. The hardening heat is higher than the critical range and depends on the shape and size.

HINTS AS TO TREATMENT.

The following table will show the approximate critical ranges and hardening temperatures of steel relative to the carbon content:

Carbon Content, %	Critical Range, F.	Hardening Temperature, F.
0.60	1,340 to 1,380°	1,400 to 1,460°
0.70	1,340 to 1,375°	1,400 to 1,450°
0.80	1,340 to 1,365°	1,390 to 1,450°
0.90	1,340 to 1.360°	1,375 to 1,450°
1.00	1,340 to 1,360°	1,375 to 1,450°
1.10	1,340 to 1,360°	1,375 to 1,430°

The forging temperature for drill steels should be from 1550 to 1600 degrees F. Do not continue the forging operation after the steel fades from a red color, as forging a steel cold or below its critical range distorts the structure and tends to produce brittleness. Reheat the steel, if necessary, to finish the forging operation.

In annealing, heat the steel just above the critical range and cool slowly. In hardening, if the steel is rapidly heated, it should be held just long enough to heat thoroughly. Uneven heating results in a soft center. On the other hand, the steel should neither be held too long at the hardening temperature nor soaked, as such treatment opens the grain or enlarges the crystals, tending to brittleness.

The quenching bath should be large and deep. Clean water is a suitable quenching medium, but for uniform results it should be kept at 60 to 80 degrees F. After the steel is quenched, it is in its hardest condition, with a tendency to be brittle; accordingly, it will not well resist shock. To obviate this, it is well to draw the temper, thus sacrificing a small amount of the original hardness for a like gain in toughness. The temperature to which the steel should be drawn will depend on the condition of the rock. This must be determined by experimenting. The drawing temperature will vary from 350 to 600 degrees F. Use a heavy or 600 degrees fire-test oil for the drawing medium, reading the temperature with a thermometer. The "Tycos" armored thermometers, of the Taylor Instrument Co., Rochester, N. Y., are suitable.

In treating the shank end of the steel, after forging and annealing, heat to the proper hardening temperature and quench in oil. The oil tank

should be large and the oil kept at a uniform temperature by circulation through a coil surrounded by water or by circulating the water through a coil around the inside of the oil tank. No. 2 soluble quenching oil, made by E. F. Houghton & Co., Philadelphia, Penn., is suitable. When the steel has reached the temperature indicated by the pyrometer, the color of the heated steel will be the same as that of the iron pipe in which the base-metal couple is in direct contact with plugged end inserted in the furnace.

In conclusion I desire to urge coöperation with the steelmaker in all drill-steel troubles, as I believe the steel-maker is as anxious to give satisfaction as the user is to obtain it.—"Engineering News."

THE MICHELL THRUST BLOCK.

The principle of the Michell bearing, no matter whether the surfaces are round as in a journal or flat as in a thrust block, consists of the creation and the maintenance of a lubricating film of such a persistence that no matter how great the bearing pressure, the metallic surfaces can never come into contact (Fig. 1). In order to obtain this, the bearing surfaces are so arranged that they are free to take up a slight inclination to each other, so that when the machinery is in motion, the oil is drawn in naturally and is free to escape at the trailing edge of the brass as well as at the sides, ordinary journal bearings do this to a certain extent already, due to the displacement of the shaft with relation to the brass as it revolves, caused by the tendency of the shaft to "climb" the bearings; but the flat surfaces of a thrust bearing present an entirely different problem.

In the well-known multi-collar marine thrust block, the areas of bearing surface are calculated on a low basis of 40 to 60 pounds per square inch bearing pressure, but in the Michell single-collar thrust block, the bearing pressure is about 10 times greater. The friction, however, is only 1-25th of what it is in the old-fashioned block, because instead of running on greasy or dry white metal the shaft is now running on an oil film so persistent that it is almost impossible to break it down. Experiments have been made which prove that with a properly designed Michell block the bearing pressure may be increased until the white metal, although remaining quite cold, actually begins to squeeze out like butter without breaking down the oil film; but it is not recommended to run thrust bearings at these intense pressures of 8,000 to 10,000 pounds per square inch, 400 to 500 pounds per square inch being the bearing pressure provided for in actual practice.

The general arrangement of a self-contained Michell thrust bearing is shown in Fig. 2, and it will be observed that the bearing collar, instead of being a continuous flat ring or horseshoe of white metal, is split up into a series of loose sectors—usually six or eight—each sector capable of a slight tipping or rocking motion due to being cut away at the back, or to being allowed to pivot on a hardened steel point. Fig. 3 is a diagrammatic representation of these loose sectors from which the tipping or rocking motion can be readily understood. An important point to notice is the position of the rocking ridge or pivot, which should always be beyond the center of the tipping block in the direction of rotation of the shaft. Another form of Michell thrust block is shown in Fig. 4, which illustrates an enclosed type of marine thrust bearing, whilst the method of applying the Michell principle to existing multi-collar thrust

*Precis of a paper read by J. Hamilton Gibson before the Liverpool Engineering Society.

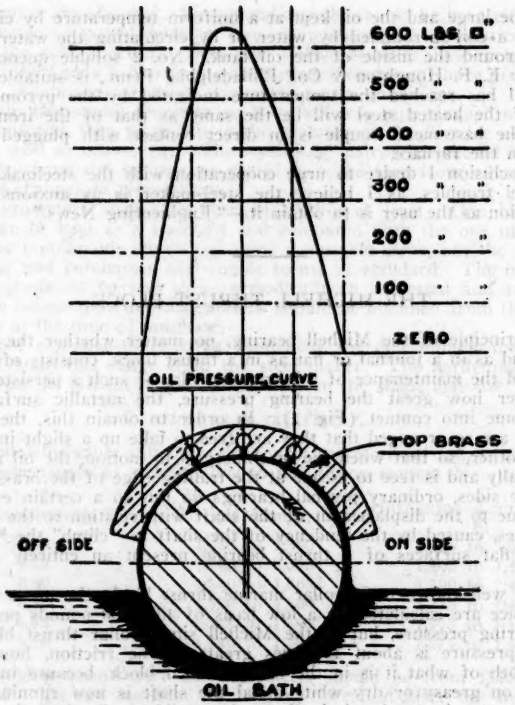

FIG 1.—ILLUSTRATING THE FORMATION OF A PRESSURE OIL FILM IN A JOURNAL BEARING.

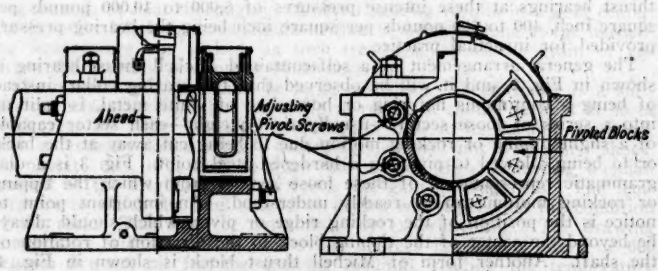

FIG. 2.—SELF-CONTAINED MICHELL MARINE THRUST BEARING.

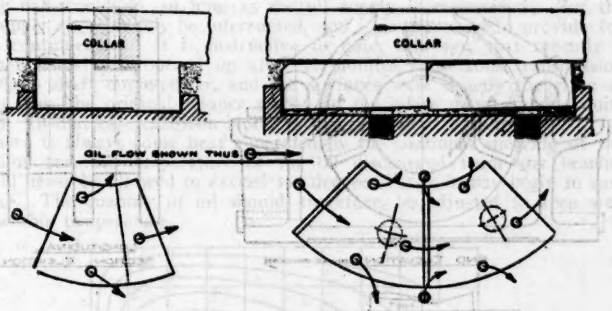

Fig. 3.

ILLUSTRATING LINE PIVOTING IN A MICHELL THRUST BLOCK. ILLUSTRATING POINT PIVOTING IN A MICHELL THRUST BLOCK.

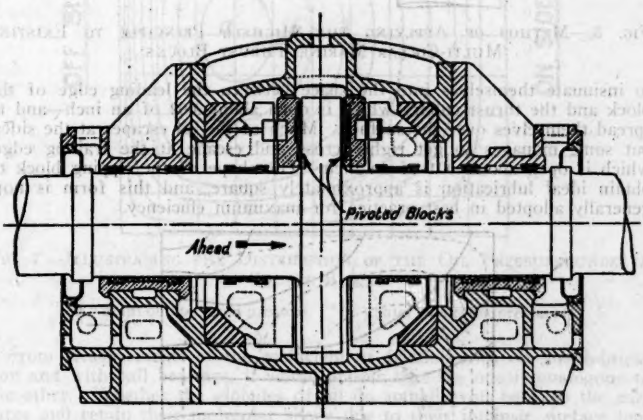

FIG. 4.—ENCLOSED TYPE OF MARINE THRUST BEARING.

blocks is shown in Fig. 5. The flow of oil across one sector of a Michell block has been demonstrated by means of experiments with the Hele-Shaw stream-line apparatus, and Fig. 6 shows the result of such an experiment using a square tipping block. The lines of flow of oil are represented by full lines on the diagram, and the lines of equal pressure by broken lines. These contour lines are shown in Fig. 7. The position of the rocking point corresponds approximately with the position of the point of maximum oil pressure (*a*) on the surface of the block; the resultant point of the varying pressures is shown at (*b*), and the couple thus formed tends to open the block at its leading edge or "on" side. The parallel streams of oil approach the block in the direction of the arrows (Fig. 6), and most of the oil is deflected round the free edge of the block. But due to the tipping of the block, streams of oil manage

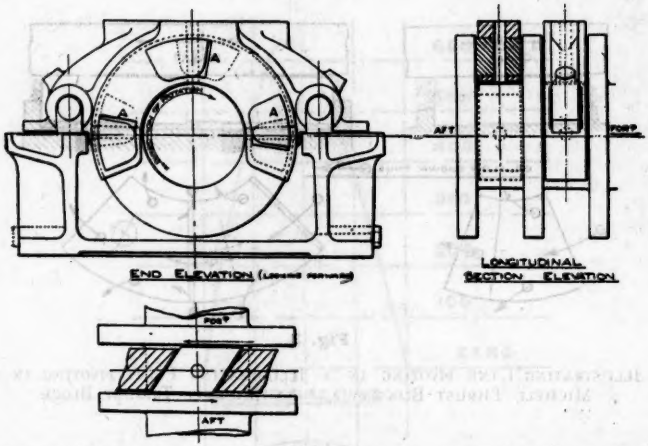

FIG. 5.—METHOD OF APPLYING THE MICHELL PRINCIPLE TO EXISTING MULTI-COLLAR MARINE THRUST BLOCKS.

to insinuate themselves into the space between the leading edge of the block and the thrust collar—which is open about .002 of an inch—and to spread themselves over the surface. Much of the oil escapes at the sides, but some manages to get right across and escape at the trailing edge, which is open about .001 of an inch. The shape of the tipping block to obtain ideal lubrication is approximately square, and this form is now generally adopted in best practice for maximum efficiency.

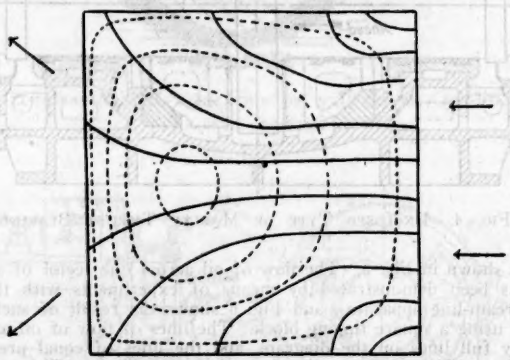

FIG. 6.—ILLUSTRATING THE DISTRIBUTION OF OIL PRESSURE UNDER A MICHELL TIPPING BLOCK.

Lines of Flow——— Lines of Equal Pressure.....

It might be argued that if the thrust surfaces never come into metallic contact, being always separated by an oil film, why is it necessary to go to the expense of white metalling the tipping blocks? There is no necessity for white metal, or even for brass, good cast-iron scraped per-

fectly flat would do, so long as the oil supply is maintained. But the oil supply may possibly be interrupted, and it is necessary to provide for such contingencies. It is instructive to note, however, that recently a Michell thrust was opened up after 12 months' continuous transmission of 20,000 shaft horse-power, and the surfaces were exactly the same as when new, the original scraper marks on the white metal looking quite fresh. In that case cast-iron blocks would have done equally well.

There is always some heat generated by the continual shearing of the oil film, and the temperature of the oil discharged from any bearing should never be allowed to exceed 150 degrees F., or it may begin to carbonize. The quantity of oil should, therefore, be adjusted to keep well below this temperature.

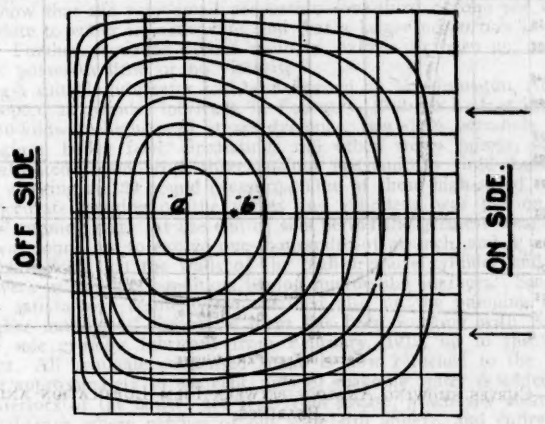

FIG. 7.—ILLUSTRATING THE DISTRIBUTION OF THE OIL PRESSURE UNDER A SQUARE BLOCK.

"A" Point of maximum pressure. "B" Center of varying pressures.

From measurements of the coefficient of friction with oil film lubrication and with ball bearings, it would appear that the one is analogous to the other. Whether the globules of oil do actually roll between the surfaces and retain their molecular shape due to their intrinsic surface tension it is difficult to say, but that some such action takes place is conceivable, especially as the character and the magnitude of the coefficients shown in Fig. 8 are so similar. The low efficiency of the ordinary thrust is strikingly shown in this diagram.

The Michell journal bearing has the top and bottom brasses split into some six separate tipping blocks, which shortens the overall length by a considerable amount and eliminates trouble due to the heating of the end of the brasses caused by the slight deflection of the shaft as a whole. —"Shipbuilding and Shipping Record."

GRAPHITE CYLINDER LUBRICATION.*

One reason which has mitigated against the economy of utilizing engine exhaust steam for various purposes is the contamination of the steam by the oil used for cylinder lubrication. Oil in steam beyond the cylin-

*From a paper by Mr. E. W. Johnston.

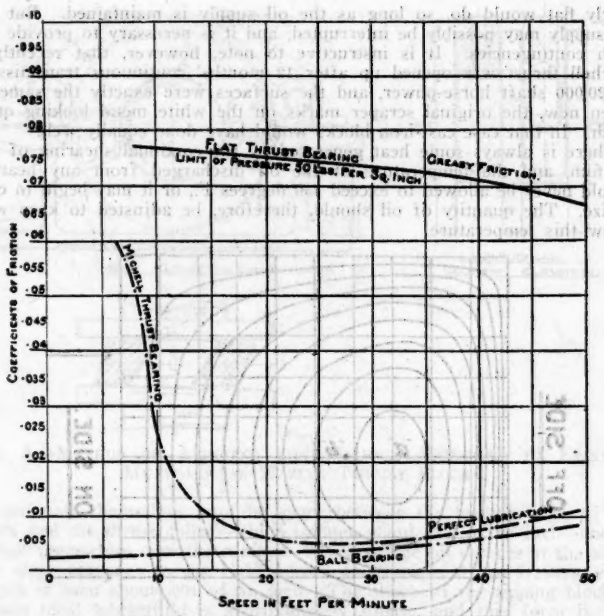

FIG. 8.—CURVES SHOWING ANALOGY BETWEEN FILM LUBRICATION AND BALL BEARINGS.

ders, or in the waters of condensation, brings only trouble to the engineer. By a process of careful filtration, combined with the employment of a coagulant, it is doubtless possible to obtain a condensate sufficiently free from oil to be fit for boiler feed purposes; but there is no system yet known capable of entirely eliminating oil from steam. The problem, then, is to find an alternative lubricant to oil. Now, graphite in certain forms is an excellent lubricant. The discovery by Dr. Acheson of artificial graphite has, in particular, given to engineers a remarkably pure product, free from trace of grit, in an unctuous form, which readily permits of its disintegration to a powder comparable in fineness with, say, fine pastry flour. Late in 1906, Dr. Acheson, in the United States, discovered that by a process of what he terms "deflocculation," he was able to carry the subdivision of graphite much beyond that attainable by ordinary means. Furthermore, he found that by mixing a weak solution of tannin with his disintegrated lubricating graphite, and agitating the mixture, the minute pieces of graphite were broken asunder. The extremely fine particles of graphite thus produced show no tendency to cohere. Let it be remembered that by the prior process of disintegration the graphite particles would pass a 40,000-mesh sieve; but to such an extent does this deflocculation or breaking down take place that each of these very small graphite particles is sub-divided into about 1,000 parts. The particles of deflocculated graphite are so small that they will pass

through the finest chemical filter paper almost as readily as pure water. Moreover, the graphite, although having a specific weight of approximately two and a-quarter times that of the water in which it is diffused, remains in perfect suspension. If, however, a few drops of acid—such as hydrochloric—or other electrolyte be added, the graphite again flocculates into masses of such size that it will no longer remain in suspension.

We have here a remarkable new lubricant, "Aquadag," chemically inert in pure water, a medium convenient for application at least for internal lubrication, and of such extreme fineness as to be capable of readily entering the pores of metals, evening up their surfaces, and forming a coating of high durability and exceptional lubricating properties. Another remarkable feature of Aquadag is that numerous and long-continued trials show that the very small proportion—one-third of one per cent.—of graphite to water is inadequate, and that a larger proportion is superfluous. Further, Aquadag cannot emulsify, oxidise, or burn up, and, unlike oil, possesses little or no viscosity.

When a suitable lubricator had been devised by Mr. Johnston, Aquadag was adopted as cylinder lubricant, in February, 1912, on a plant including three 50-kilowatt high-speed steam dynamos, two deep bore-hole pumping engines, boiler feed, circulating, and other steam pumps, supplied with saturated steam from three 30-foot mechanically fired Lancashire boilers working at 120 pound pressure. One of these high-speed engines, after accurate gauging of the valves and cylinders, was put on a six months' running test. At the end of this period the greatest wear at any point was found not to exceed one-thousandth of an inch, and it was particularly noticed that the walls of the high-pressure cylinder and piston rings were in faultless condition, having mirror-like surfaces. Since uniformly satisfactory results were obtained also on the pumping engines and other auxiliaries, the entire plant has been working with Aquadag as the sole cylinder lubricant from February, 1912, up to the present moment. All available condensed steam is now returned to the boilers, so that approximately 10 per cent. only of make-up water is added daily. The interiors of the boilers are free from grease, practically clean down to metal, save where patches of old scale still adhere, and entirely free from suspicion of "pitting." Formerly a considerable amount of scale of very tenacious character had to be dealt with, which resulted in loss of efficiency, extra labor, and wear and tear to the boiler. Now, on the other hand, a feed water of considerably higher temperature and exceptional purity is available, resulting in further considerable fuel economy. After almost five years' daily use, there is found to be no deposit throughout the receivers, ports, valves, and cylinders in excess of the thin coating formed in the first few weeks of use, and little, if any trace in the exhaust pipes, condensers, or other tracts beyond the engines. This paper is founded upon experience gained in the employment of Aquadag in connection with engines running on non-superheated steam, and the writer is not entitled to speak as to results which might be derived in conjunction with superheated steam, although he has reason to suppose that in one instance, at least, his system is giving satisfactory results where some degree of superheat obtains. There is, however, no question as to the ability of deflocculated graphite to afford efficient lubrication; but practical difficulties in the way of "atomising" it into the current of steam yet remain. Experiments made by the writer, however, lead to the conclusion that the problem is capable of solution. In the use of Aquadag for the cylinder lubrication of internal combustion engines, many interesting problems are involved in the elucidation of which the writer hopes it may be his privilege to assist.—"The Engineer."

BOOK REVIEWS.

STEAM TURBINES. By WILLIAM J. GOUDIE, B. Sc. Size. 5½ by 8½ inches. Pages, 520. Illustrations, 230. LONGMANS, GREEN & COMPANY, 4th Ave. & 30th St., New York. Price $4.00 net.

The first portion of this book is devoted to detailed descriptions of commercial types of steam turbines now on the market, while the second portion is devoted to what may be termed the technical part of the subject; that is, the mathematical discussion of the action of steam in the turbines and the determination of the size and shape of the various parts. While the book has been written primarily to suit the requirements of engineering students, it will also be found useful by engineers who have to deal with the design and operation of turbines, for the reason that the production of the commercially successful turbine depends to a considerable extent on the use of design coefficients derived from practice, and in this book these have been applied to suit the conditions of each particular case. The fundamental principles underlying any scheme of design are clearly defined and the practical limitations to the application of theory which experience has shown to be necessary in successful turbine work are stated. By the introduction of practical coefficients and data quoted from various reliable sources, the author has endeavored to place a practical system of calculation before the reader. The matter is arranged progressively, so that the reader may work through the detailed calculations relating to the various portions of the machine in a logical order, finally ending with a general method of calculation for the provisional determination of the general proportions of any given type to fulfill specific conditions of operation.

PRACTICAL MARINE ENGINEERING, DURAND. Fourth Edition, Revised and Enlarged by CAPTAIN C. W. DYSON, United States Navy. Size 6 by 9 inches. Pages, 982. Illustrations,

550. Price $6.00 net. MARINE ENGINEERING, 461 8th Avenue, New York City.

The excellent treatise of Professor W. F. Durand which gave such a large amount of useful information on those parts of the general field of Engineering which are of special interest to the practical marine engineer has been brought up to date in this revision. In addition, the third edition of this book, which consisted of three separate and distinct parts, written at different times, has been so rearranged as to bring these parts together to form one connected book of fifteen chapters.

It is an addition to marine engineering literature of the greatest value.

FUNDAMENTALS OF NAVAL SERVICE. By COMMANDER YATES STIRLING, U. S. Navy. Size 4½ by 7 inches. Pages, 575. Illustrated. Price, $2.00 net. J. B. LIPPINCOTT, Philadelphia.

A book on the Navy prepared for use as a Manual to those who may choose to serve their country on the water but of general interest to those who want to know something of the Navy.

In a single volume there is given an insight into the workings of all parts of every type of ship, and the functions and history of every type of ship. By means of charts and illustrations the three great variables of fighting ship construction—speed, armor and gun-power—are treated in a way to give an intelligent grasp of the greatest strategical problems. The part each ship, whether it be a battle cruiser, or a collier, plays in a battle of the fleet upon the high seas; the method of tying knots; the fundamentals of navigation; the organization of the administrative work of the Navy Department, and the organization and purpose of the Naval War College, and of every type of naval base and battleship; the importance of sea power as developed by Admiral Mahan, all methods of signaling, and the part that electricity, engineering, etc., play in the Navy—are all items giving an idea of the encyclopedic nature of the volume.

ASSOCIATION NOTES.

The following members and associates have joined the Society since the publication of the last JOURNAL:

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Maxwell, William L., 863 Bush Street, San Francisco, Cal.

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